MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS



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This Month's Cover: Eastman Sixteen-20 projector has pushbutton control and enclosed, flexible-shaft drive. Optical system has magnesium-fluoride coated lenses, increasing transmitted light power. A heat filter drops automatically in place if camera drive slows down, protecting film from heat of lamp.

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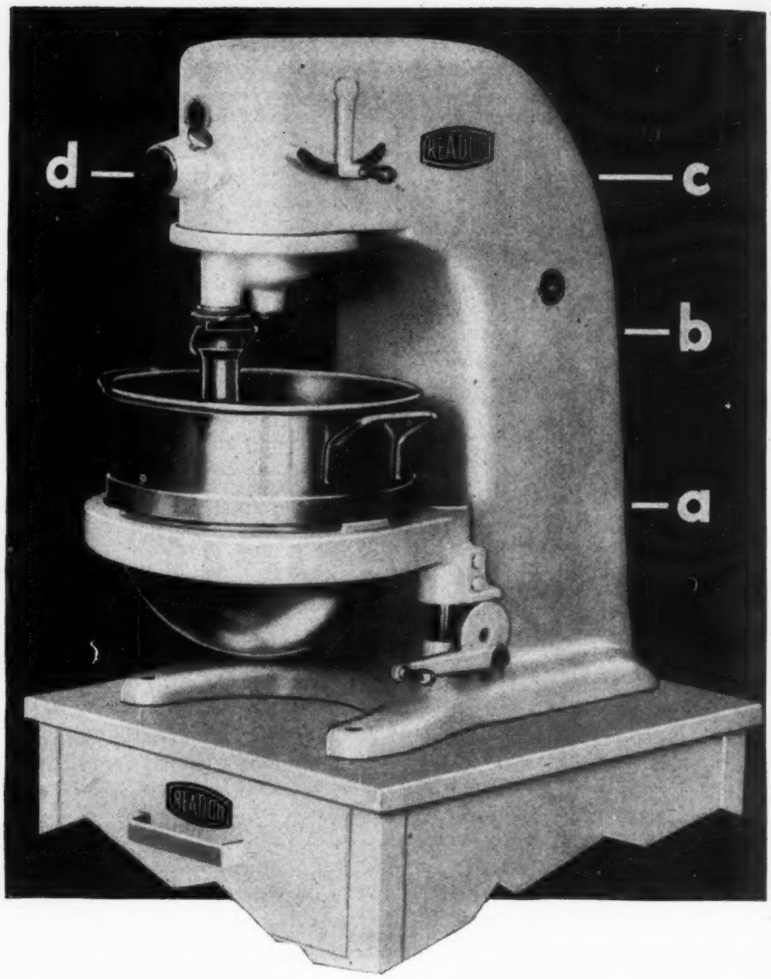
February, 1948    Vol. 20, Number 2

**OHIO GENERAL PURPOSE MOTORS** range from 1/30 to ¾ H. P. — for split phase, polyphase, capacitor and direct current operation.




# A good mixer makes a lot of dough

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# Itemized Index

**Classified for convenience when studying specific design problems**



**MACHINE DESIGN is indexed in Industrial Arts Index and Engineering Index Service, both available in libraries generally.**

# Digest

**TIMELY HIGHLIGHTS ON (GE) PRODUCTS**

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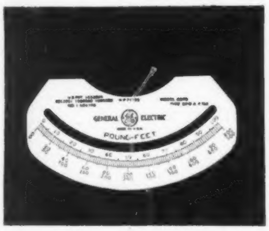


Here's a torque meter that gives you quick, accurate measurements of torque directly in pound-feet without absorbing power from the prime mover or load. You simply couple the shaft unit between the load and the driving unit. No cradling is necessary. Absolute accuracy within ± 1% of full-scale reading is provided—repetitive accuracy within ± 0.5%. Meter consists of shaft unit, power unit, oscillator and indicating instruments. Check Bulletin GEA-4441.

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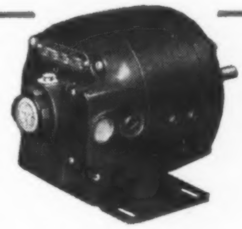

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# Topics

SYNTHETIC RUBBER plants should remain in the Government's hands until enough natural rubber has been stockpiled to make the nation secure in the event of another emergency, according to statements issued by The Goodyear Tire & Rubber Co. and the United States Rubber Co. Sale at this time, as contemplated, might delay stockpiling of natural rubber during the period of plant transfer.

FIRST COMMERCIAL application of hydraulic torque converters to passenger cars is used on the 1948 Buick Roadmaster as optional equipment. Except for an emergency low and a reverse gear, the transmission requires no manual control. Ratio is infinitely variable to fit the power characteristics of the engine.

DIFFRACTION GRATINGS for spectrographic analysis are increasingly becoming more important as a substitute for prisms of optical quartz which is virtually unobtainable today. These gratings have 15,000 straight, parallel lines per inch, each of which must be accurate to better than one-millionth inch.

WIDER USE of bearings made to uniform measurement will enable producers to take fuller advantage of standard tools, gages, fixtures and raw materials and to effect economies that can be passed along to the consumers. Tailor-made bearings, according to A. S. Murray, SKF Industries Inc., often cause acute supply situations by making it more difficult for the ultimate consumer to obtain replacements.

MIDGET AUTOMATIC PILOT has been developed experimentally at Westinghouse. Weighing only 35 pounds the autopilot is expected to be applicable to light planes, keeping positive control even throughout loops and barrel rolls.

SINTERED ALUMINA, having great mechanical strength and hardness, may prove useful in the manufacture of blades for gas turbines. Also, it as well as other sintered oxides are used successfully for cutting tools. For instance, hard rubber plate has been machined with a sintered ruby tool. Other ceramics have been used for cutting aluminum and silumin.

OXIDE COATINGS on aluminum, obtained by simple immersion method developed by Colonial Alloys Co., protect the surface and maintain its natural color. Aluminum that has been mechanically or electrolytically etched or chemically polished, as well as in the as-furnished condition, can be treated without noticeably changing the appearance.

SARAN has become a descriptive name of a plastic material in the interest of simplifying identification of thermoplastic resins chemically known as vinylidene chloride copolymers originally developed by The Dow Chemical Co. All trademark rights have been formally released.

SYNTHETIC RUBBER developments include three improvements: One makes the synthetic nonstaining and odorless through the use of a new rubber stabilizer or preservative. The second gives GR-S low water-absorption quality, achieved through the elimination of salt in its manufacture. Thirdly, a softer rubber has been developed by the use of a cross-linking agent in compounding, improving appearance and production efficiency.

FOUR MAGNETIC MATERIALS made by the Japanese have not been made domestically. These, according to reports of the Department of Commerce, Office of Technical Service, include: Alfer, an iron-aluminum alloy developed as a substitute for nickel; Sendust, a high permeability alloy of iron, silicon and aluminum; and two permanent magnet materials, NKS and OP developed before the war.

# Designing for Economical Manufacture

By Benjamin N. Ashton
*President*
*Electrol Inc.*
*Kingston, N. Y.*

*CURRENT HIGH labor and material cost necessitate closer appraisal of designs from a processing standpoint. In this article, B. N. Ashton outlines a method which has resulted in substantial cost reduction in manufacture. Improved performance as well as simplified manufacturing are the concrete result of giving production methods adequate consideration during design*

IF production processes are given their due attention right on the drawing board, *Fig.* 1, the designer of machines and machine parts can reap many benefits. All too often in the past engineers have worked with but one main idea, the meeting of technical problems; too

*Fig. 1—Keeping closely in mind the production method by which his parts can be most easily produced, the designer is assured of lowest cost and negligible production problems*

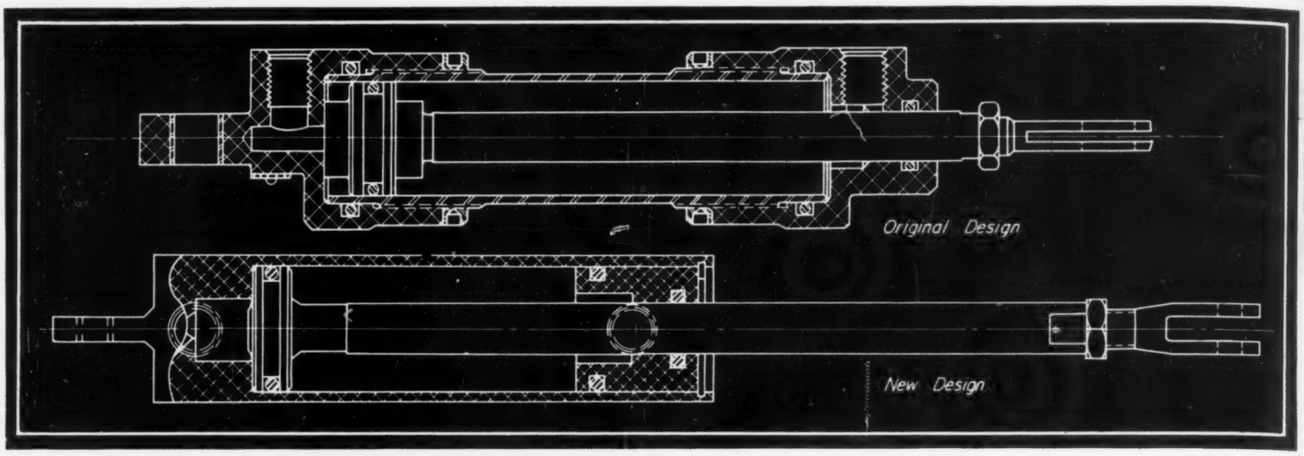
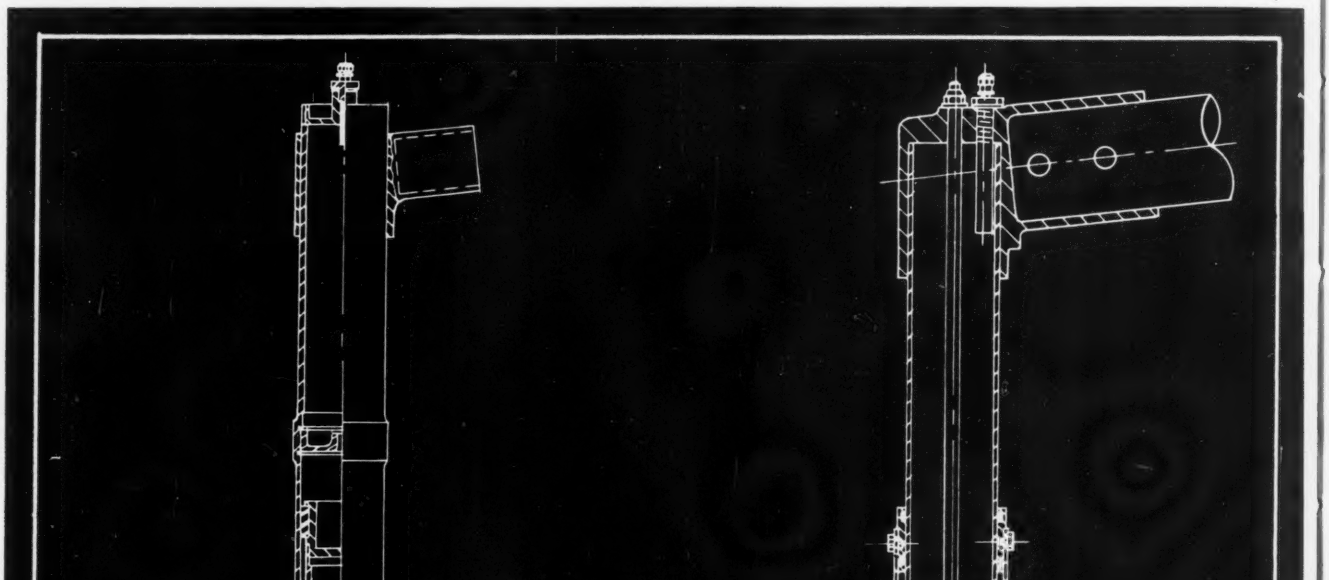



*Fig. 2—Above—Conventional actuating cylinder for tail wheel extension and retraction and new, simplified design which cost but one-sixth as much*

*Fig. 3—Below—Conventional landing gear oleo strut and new design which incorporates fewer and greaty simplified machining operations*



seldom has anywhere near equal weight been given to the problems of manufacturing.

The designer who, in meeting his technical problems, keeps in the forefront of his mind the practical requirements of manufacturing can, invariably, design a product costing only a fraction of that which would result if manufacturing did not receive its proper attention. In addition, his design usually will be simpler, lighter and often better technically. Although examples cited in this article are hydraulic items, because they are the products which we manufacture, the principles discussed can be applied just as easily to any fabricated parts with equal results.

While cost is the greatest single benefit resulting from simplified design, we have had enough experience with simplification of aeronautical and industrial parts to demonstrate conclusively that other benefits are inherent. For example, the basic design of the Electrol 500 series hydraulic cylinders was evolved from a much more complex series manufactured during the war. The new cylinders are lighter and simpler, *Fig.* 2, yet do the same job. In one typical instance cost has been cut to approximately one-sixth, the entire reduction being due to design changes since, rather obviously, labor and other costs have not dropped.

## Designer Must Know Processes

Achievement of reductions like this can come only where the design engineer is not only familiar with the general manufacturing processes but also has intimate knowledge of the equipment and facilities in his factory and has been thoroughly indoctrinated with the shop practices in his plant. He must know from experience the capabilities of the automatic machinery in the factory so that he can design his product to make optimum use of this equipment. He must know what stock supplies can be fitted to his task and whether or not they can be purchased from outside sources for less than they can be made in his shop. He must have full knowledge of other items produced in his factory so that he may use, wherever possible, parts already in production to take advantage of the savings resulting from the manufacture of larger quantities.

Besides his knowledge of his factory's capabilities, the designer must keep abreast of new developments in many allied fields and must not be hidebound by convention, especially with regard to use of new materials. He must keep in the forefront the problems of manufacturing, giving them equal weight in his considerations.

Much of our success in designing for simplified manufacturing is due to close adherence to these principles. We keep our engineers posted on our shop facilities and, when new equipment is installed, each of them has a chance to study its features. They themselves work in an office adjoining the shop floor and the shop men have access to the design room. Their proximity to the shop tends to keep the designers cognizant of shop problems. Since, whenever practical, the man who designs an item supervises its production during the early stages, our designers learn from actual experience how to make most efficient use of our equipment. In addition, this close liaison with the shop often results in further simplification of the item and in ideas later incorporated in other units.

We expect our design engineers to keep abreast of new developments in the various metalworking fields as well as in their particular hydraulic field so that they may make good use of new methods of fabricating parts and new techniques in manufacturing. To help them keep abreast, we maintain a

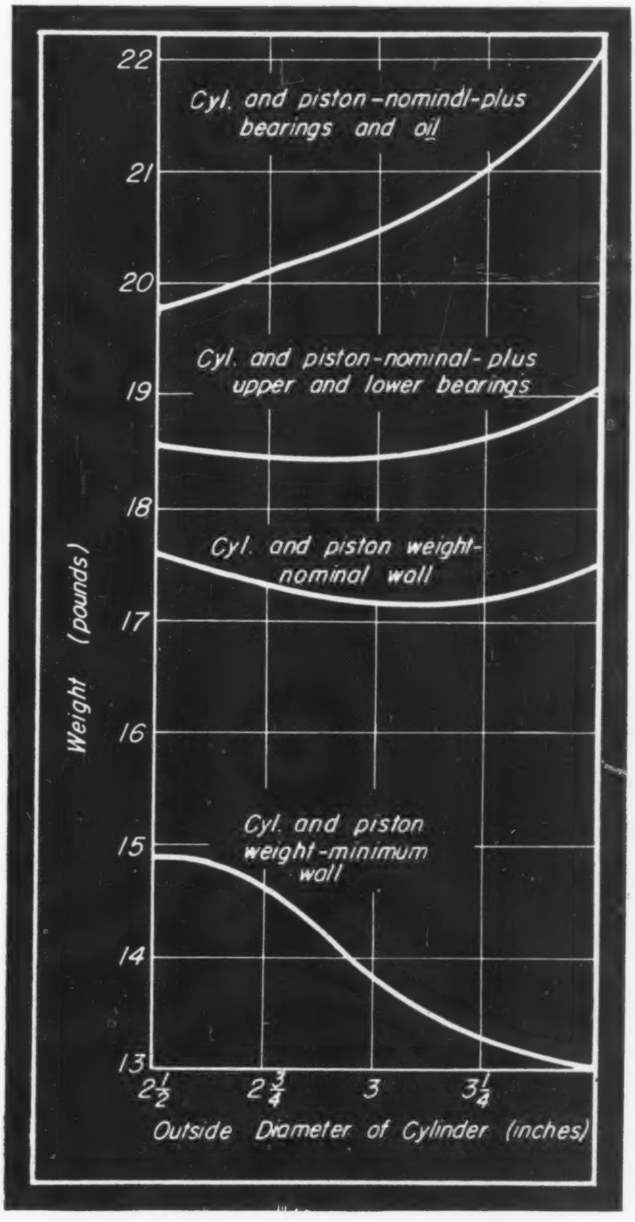



*Fig. 4—Chart of oleo strut assembly weight vs cylinder OD upon which most favorable cylinder size was based*

ready reference library in the design room. And an engineer, not a stenographer, supervises it and keeps it up to date.

It should be borne in mind in discussing simplified design that while, generally, a simplified design has fewer parts this is not always the case. Ease of fabrication may sometimes dictate use of

two or more parts to replace a single complex piece. For an example, we have developed simplified hydraulic landing gears for aircraft, *Fig.* 3, which have no fewer parts than the average oleo gear and have more parts than some. However, the gears are manufactured at far lower costs than other landing gears and have weight and strength advantages as well. These landing gears were designed expressly for the purpose of bringing the many advantages of oleo gears into the lightplane field and in order to make them feasible cost-wise, it was essential to simplify the manufacturing processes.

The most important savings which result from simplified design of hydraulic items, we have found, come from the reduction in the number of operations required for fabricating. Next in importance is the reduction in the number of parts. Important but smaller cost reductions result from widespread use of stock parts and interchangeable parts made in the shop. It is only logical to hold the number of special parts to a minimum so that economies resulting from quantity production, especially of small parts, can be realized.

A good example of how these basic principles of simplified design have been applied to reduce the cost of fabricated items is the aforementioned actuating cylinder, *Fig.* 2. The number of operations required to produce the parts of this cylinder was reduced from a total of 45 to 21. The operations themselves were simplified and greater use of automatics was made. The number of parts was reduced from 14 to 10. The resultant cylinder was 40 per cent lighter, yet did the same job, had the same capacity and operated at the same maximum pressure, 1500 psi. Cost, as mentioned before, was reduced to about one-sixth, from well over $40 to less than $7.

To begin with, operations were designed to fit our existing machinery, the same machinery which has been used, in the main, to produce the old-style cylinders; no new capital expenditures were, therefore, necessary. The savings achieved resulted principally from the redesigning and, to a lesser degree, from the savings in materials made possible by the new, lighter design.

## Impact Extrusion Utilized

Simplification of the cylinders began with the major pieces, the body, head and end, and the piston and rod. An impact extrusion replaced the old body, in one step eliminating several operations by making the body and head one piece.

Operations on the body and head were reduced to six: (1) bore, recess and face, (2) straddle mill and burr, (3) drill, spotface and tap, (4) burr and degrease, (5) anodize, and (6) inspect. The old-style cylinder required 12 operations on the body alone and another 8 on the head. Thus the operations saved on the body and head alone numbered 14.

The gland end of the new model required only another six operations: (1) drill, face, recess, ream and cut off (all done on automatics), (2) turn groove, (3) drill, countersink and tap, (4) burr and degrease, (5) anodize and (6) inspect. The gland end of the previous model required 10 operations.

Reduction of operations and manufacturing costs on the piston and rod were comparable. An upset forging replaced the old rod and piston, formerly made as two pieces. The new rod and piston required only nine operations: (1) centerless grind,

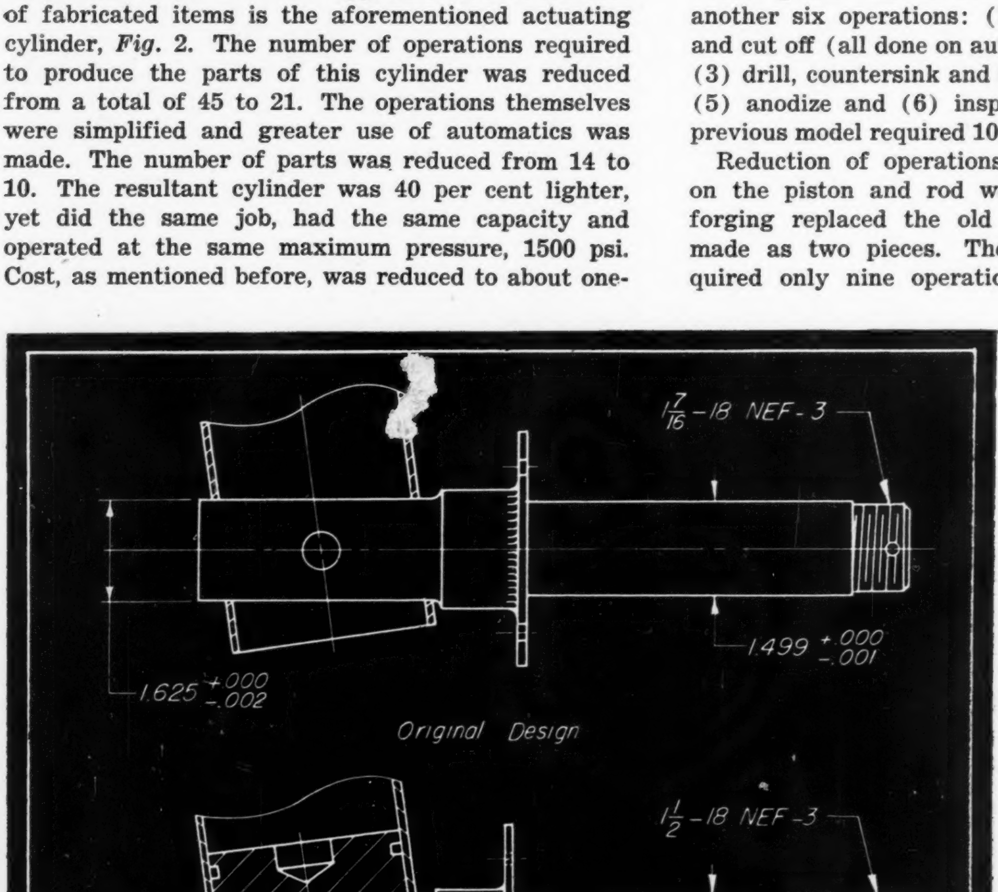



*Fig. 5 — Left—Original landing gear axle with welded flange required 0.77 man-hours to machine while the new design, in two pieces, required but 0.44 man-hours to complete*

(2) turn groove, chamfer and face piston, (3) drill, face and countersink rod, (4) straddle mill, (5) grind O.D., (6) chrome plate, (7) centerless grind, (8) polish, and (9) inspect. Heretofore, the piston blank required three operations and the piston rod three operations before they were assembled. After assembly they required an additional 12 operations, for a total of 18—just twice as many as the new piston and rod require.

Proof that the principles of simplified design are not limited in their application can be found in many other Electrol hydraulic products. With the Electrol unloader valve, for example, the old-style valve, Model No. 220, required 65 operations and only two parts were partially made on automatics. Yet it was directly competitive in price and quality with similar valves made by other concerns. Redesigned for simplified manufacturing, the valve became Model No. 465, which requires only 54 operations and all eight parts are made almost entirely on automatics. In addition, the following pieces were eliminated: Two packings, three special washers, a special screw, three special nuts, two special seals and two precision ball bearings.

## Simplification Applied to New Designs

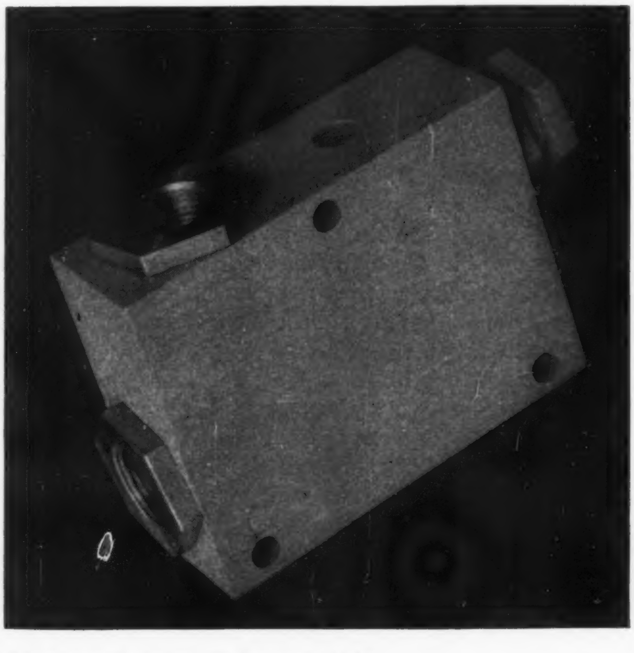
Reduction in the work necessary to produce the new model valve actually was greater than is indicated by the reduction in the number of operations from 65 to 54. On the body of the Model 220, for instance, one operation called for the milling of five sides while on the Model 465 valve, the same operation requires milling of only two sides. Savings like this were achieved on other parts as well.

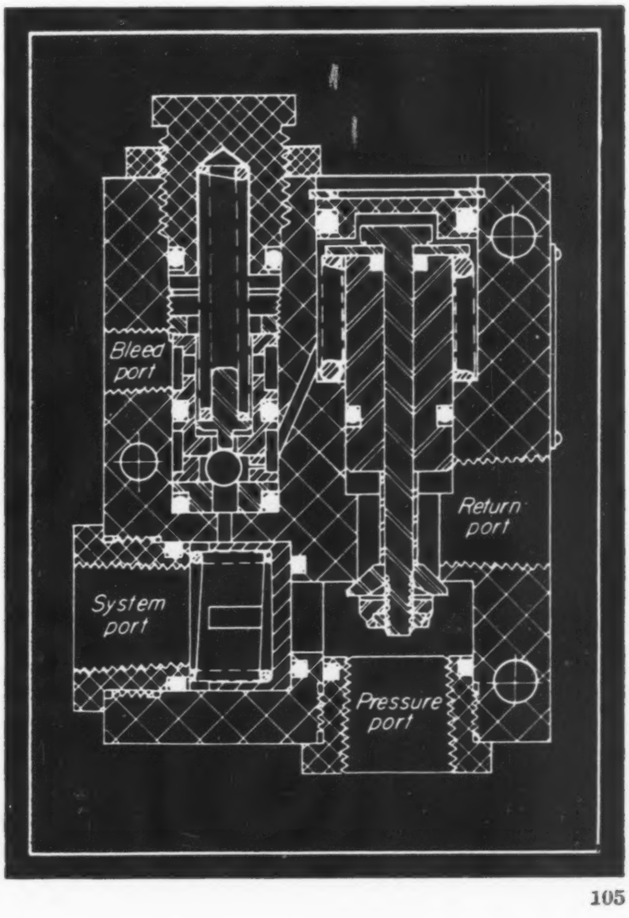
To this point only *re*designing for simplified manufacturing has been discussed. The principles expounded here work just as effectively when applied to original designs our experience has shown. We have applied them successfully to articles which we did not manufacture previously, starting our production of them with simplified designs. One example is the Electrol series of hydraulic landing gears mentioned earlier, *Fig.* 3. Since an aircraft landing gear is a relatively complex item which must withstand hard usage, often under severe conditions, and which poses difficult weight control problems, it would be well to trace the steps in simplified design which led to the successful production of this series of landing gears—especially since our cost reduction, achieved through simplified manufacturing, brought into the lightplane field for the first time the proved advantages of oleo gears at prices which the industry could afford.

Experience with aircraft hydraulic equipment had convinced us that there was a good market for efficient oleo landing gears in the personal aircraft field, but no available gears could meet cost requirements. Our market survey showed that a sufficient volume could be obtained if landing gears usable on more than one manufacturer's aircraft could be designed. Therefore, we devoted our attention to the designing of a series of standardized gears suitable for use on any personal aircraft up to 3000 pounds gross weight.

Main gear models were so designed that they could be used with fixed or retractable installations on tricycle or conventional aircraft and on amphibians and helicopters. Nose wheel gears were designed for either steerable or free-castering installations. In addition to designing for simplified manufacturing

*Fig. 6—Electrol hydraulic cut-out valve, below, and cross-sectional view, right, showing design for low-cost simplified manufacturing*





so that the cost of the gears would fall within reasonable limits, we more or less had to design for universal application. We found, in our engineering studies, that simplified design was a help rather than a hindrance in achieving this universality. It was also found possible to keep weights well within reasonable limits. For instance, each main gear assembly for a tricycle installation for a 2000-pound airplane weighs but 11 pounds and but 16 pounds for a 3000-pound airplane, axle included. The weight of assemblies for conventional installations is even lower.

## Co-ordinated Engineering Required

The simplicity of design is readily apparent in the accompanying illustrations. Close co-ordination between process engineering and design engineering was necessary to arrive at a landing gear, such as is illustrated, that can be manufactured at low cost. In determining basic design, extreme care was taken to assure that weight control would not suffer from the compromises necessary in the detail design for simplified production. Weight usually suffers severely when a mechanism such as a landing gear is designed for production only.

It was obvious from the start that complete standardization of landing gears for the lightplanes of various manufacturers was not possible. However, we found that many of the major components such as axles, scissor links, brake flanges, and oleo bearings could be made common to virtually all installations. In addition, other major parts could be made standard for several different installations so that virtually all components could be used on a multiple of installations.

A purely theoretical study of structural requirements showed that, as the diameter of the strut was increased, weight was reduced, *Fig.* 4. However, further considerations showed that as such items as attachments, bearings, spacers, etc., are added, the most favorable diameter shifted toward the small side. Then inclusion of the hydraulic oil, the factor most often overlooked, showed that the lightest gear would actually be that using the smallest practical diameter strut. Although the conclusion undoubtedly would be challenged, a careful check showed us that it could be substantiated and our design proceeded upon this basis.

After settling such details as length of stroke needed to meet requirements without imposing undue loads on the airplane structure, our design studies moved to a consideration of the choice of materials for the landing gear—an extremely important factor since material costs were to represent approximately 50 per cent of the total cost of the gear.

Stress and other requirements dictated use of steel for the cylinder, piston and axle. Consideration of practices showed that savings in material costs could be effected by using heat-treated stock with a thinner wall since the higher heat treatment did not incur increased machining time on the axle and cylinder, which are ground on the outside only.

Investigation of materials for such parts as scissor links, attachment fittings and axle socket showed that aluminum alloy forgings would cost about 15 per cent more than steel for these parts, but that they could be machined in about one-third the time required to machine a steel forging, a saving which offset considerably the additional initial cost. Use of aluminum here also resulted in a lighter unit.

One of the fundamentals in designing for simplified manufacturing is aptly illustrated in *Fig.* 3, which shows a good, conventional-type landing gear oleo design, with all of its intricate counterbores, turning operations and multiple diameters, and the simplified design. Note that in Electrol's simplified gear all machining operations are confined to either the inside or the outside of a part; machining is not necessary on two sides. Note, too, how the piston can be centerless ground and how it can then be completed merely by a flash chrome operation. This design also shows another important aspect of simplified design in the use of a simple tie rod through the center, eliminating the conventional threaded and bolted joints.

Illustrative of how continued attention to manufacturing details pays off is our experience with the axle on this landing gear, *Fig.* 5. The original axle required 0.77 man-hours to manufacture. The brake flange was welded directly to it and, following convention, the thread was one-sixteenth of an inch smaller than the basic axle diameter. Redesigned, the part is made in 0.44 man-hours, the main differences being that the axle can now be centerless ground, the brake flange can be drilled easily in a drill jig and a whole operation is saved by making the threads on the end of the axle to the same diameter as the axle itself. These were simple changes, but once again they resulted in sizable savings over conventional design.

## Method of Attack Varies

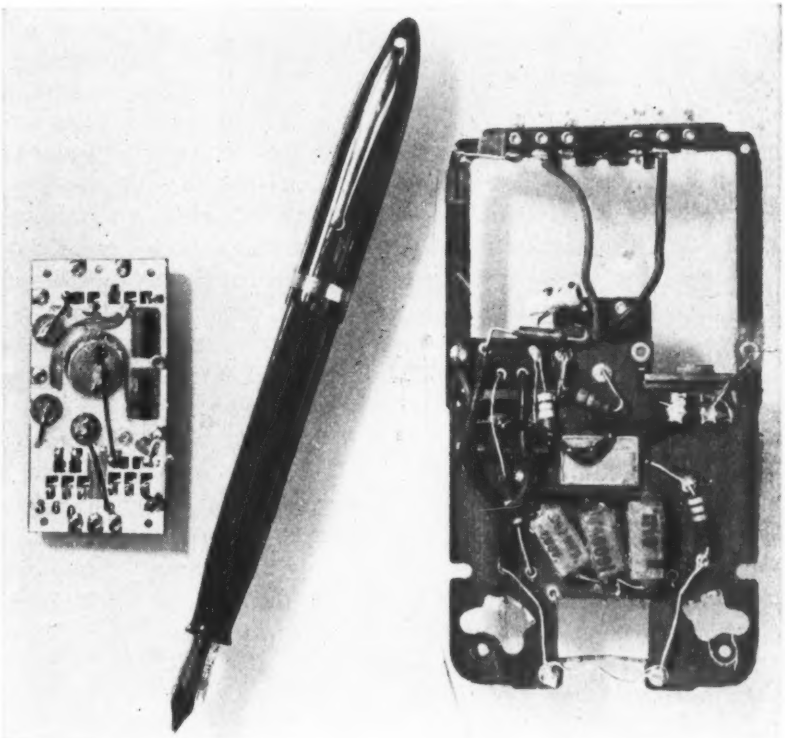
Our simplification program on landing gears, unlike that of airframe manufacturers and others, was not based upon the reduction in the number of parts in the assembly. Rather, it was based upon the breaking down of the component parts into parts which could readily be made on automatic machinery and a reduction in the number of operations—a reduction both in the number of operations performed on individual parts and on the assembly as a whole.

While the landing gear design has been discussed here in some detail, it is not the only new product to which we have applied these principles of simplification. They have been used with equal benefits on other items, *Fig.* 6, greatly varied in nature and, as was shown by the items first described, have resulted in rather astonishing savings in the production costs of redesigned articles.

Visitors at our plant in Kingston have, during discussions of our simplification program, remarked that the steps we have taken are "obvious". We agree. They are obvious and they also are "simple". That is why they have been successful. Designing for simplified manufacturing is nothing more than good, common engineering "horse sense". But it is the kind of common sense all too often overwhelmed by other design problems.

# Scanning THE FIELD for Ideas

**Printed electronic circuits**, developed during the war for use in the proximity fuse, have been utilized by Allen-Howe Electronics Corp. to reduce the size of hearing aids and improve their performance. Shown in the illustration, at right, is a 3-stage printed circuit amplifier. For size comparison a conventional 2-stage unit is included in the photograph together with a fountain pen. The side shown for each unit is that containing the largest number of condensers or capacitors. Both sides of each panel, however, carry their full share of parts.



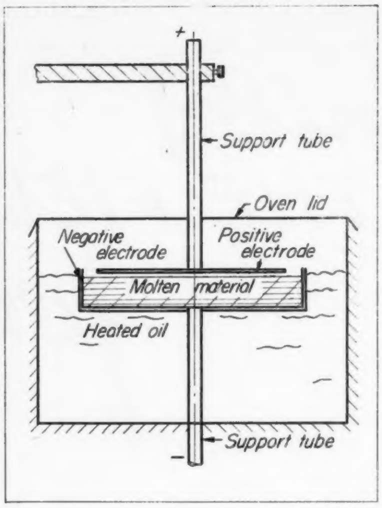
Panels are 2¼ by 1-5/32 by 3/32-inch wafers of steatite, a dense ceramic, on which have been printed and bonded the wiring and resistors. Minuscule capacitors, consisting of paper-thin ceramic disks silvered on both sides, and subminiature tubes are soldered to the circuit. Tests indicate that printed-circuit amplifiers have superior power gain characteristics over conventional units. With 1 millivolt input a unit delivers an output of 1.3 volts compared with 1 volt for conventionally wired devices. This is due to the fact that the "coupling" between output and input, producing a degenerating effect, is reduced in the simplified arrangement possible by printing.

In the production of printed circuits, silver ink is brushed over a silk-screen stencil on the steatite chassis. The resulting pattern is a group of silver lines for the exact circuit desired. The ink consists of finely divided silver or silver oxide mixed with a binder and thinned with a solvent such as acetone. After application of the circuit pattern, the chassis is heated to a temperature between 1300 F and 1500 F, bonding the silver to the steatite.

Then resistors are painted or sprayed through a second stencil in such a way that they are applied in their exact locations in the circuit. This paint is composed of a conducting material such as powdered graphite and a nonconducting material such as mineralite or powdered mica. The chassis is again baked at 300 F to cure the resistors and stabilize their characteristics.

Next, disk-like condensers are applied direct to the plate. They measure from ⅛ to ¼-inch in diameter, 0.04-inch thick. With the addition of the subminiature vacuum tubes, transformer, batteries and accessories, the amplifier is complete. The whole assembly is housed in a case about the size of a cigarette package.

**Plastic electrets**, according to available data, may replace expensive cobalt-steel magnets in those units whose functions are dependent on mag-




netic characteristics. These electrets are made from a dielectric material in such a way that they retain an electric moment or electrical polarization. Considered as the electrical equivalent of permanent magnets, electrets support the theory that electricity and magnetism are an indivisible pair.

Process for producing electrets is illustrated in the photograph and in the schematic drawing, above. It necessitates the use of a resistance thermometer, a phototube, a relay system and a field strength of about 3000 volts per centimeter. A thermostatically regulated oil bath permits control of the time interval required.

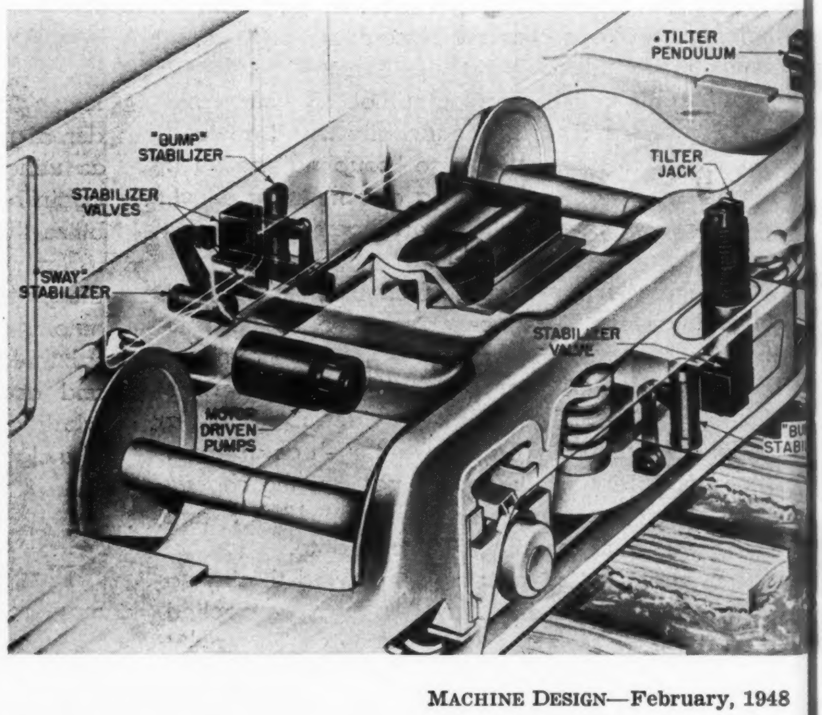
Tests reported by Plastics Research Co. reveal that both thermosetting and thermoplastic materials can be used to produce electrets, providing their molecules form unattached bonds in the molten state when they are subjected to a strong electric field and remain in this molecular arrangement after solidification and removal of the field. Among the thermosetting materials, unsaturated polyesters appear to have the most desirable features for the manufacture of electrets. These polyesters have interlinked molecular structures which permit them to react like thermoplastics until the thermohardened condition prevails, are polymerizable at various temperatures in long or short periods of time, and possess excellent physical properties in the cured state. Suitable synthetic thermoplastics include acrylic, ethyl cellulose, styrene, and vinyl polymers or copolymers.

When produced on a large scale, the essential advantage of electrets will undoubtedly be economical. Their characteristics indicate possible use in electrical measuring instruments, in electronic equipment such as vacuum tubes and microphones, in electric motors, and in timing mechanisms. Although first proposed by Michael Farady in 1839, the first electrets were produced in 1925 in Japan, using wax as the dielectric material. Such electrets were reputedly used during the war.

**Shock absorber,** shown in phantom drawing below, smooths out the bumps in railroad tracks insofar as riding comfort is concerned. Developed by Westinghouse research engineers this stabilizer is designed to eliminate 60 per cent of the bumps and sway caused by track irregularities and to enable trains to take curves at 25 per cent greater speed.

Floating weights respond to up-and-down as well as side motion and a pendulum senses the pull of centrifugal force and gravity, correcting the movement before it is felt by passengers. When the car comes to a bump or a dip, the floating weight senses the change in motion and operates valves, causing oil under high pressure to flow into the proper cylinder. The piston in the cylinder moves with just enough force to hold the car body in position. The same type of action causes relative movement between the body and wheels to correct for sidesway.

When the train enters a curve, the gyro-controlled



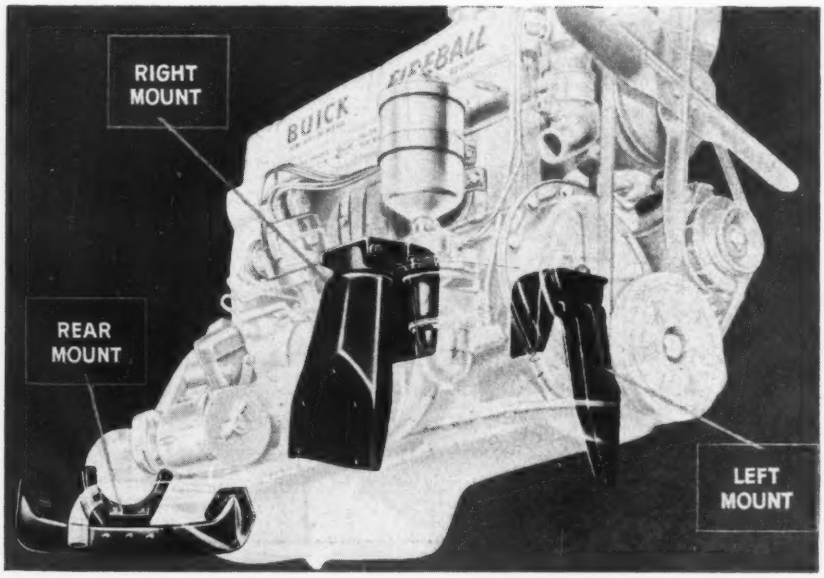
pendulum regulates two electrically driven screw jacks placed at diagonal ends of the car body. If the speed is not right for the bank of the track, centrifugal force swings the pendulum. The jacks immediately tilt the car body to the correct bank angle. This tilt mechanism can add up to 6 degrees additional bank in either direction within 2 seconds and within 1 degree of perfect equilibrium. This stabilizer limits resonance to less than one inch, an improvement of 300 per cent over other shock absorbers.

**Quality rating** of assembled ball bearings is quickly and simply obtained with the instrument shown at right. It is calibrated in terms of the height in microinches and the frequency of occurrence of radial movements of the outer bearing race as the inner race rotates. These deviations are caused primarily by waviness and similar surface irregularities of the balls and races.

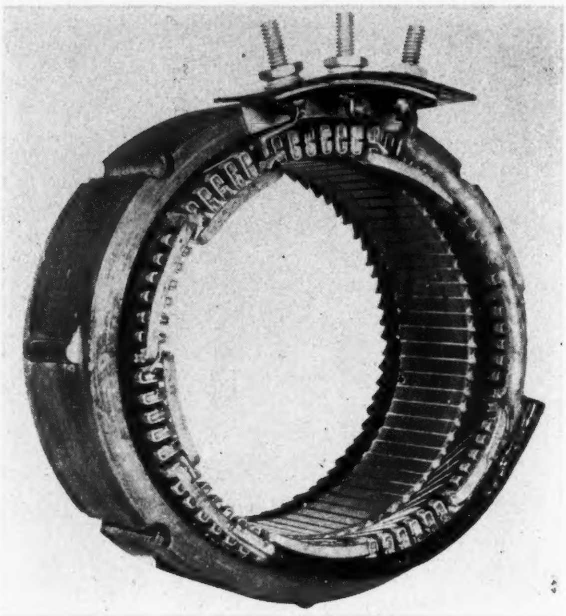
The instrument, designed by Physicists Research Co., has an interchangeable motor-driven cartridge-type spindle unit to fit the size of bearing being tested. An electromagnetic converter is adjustable on a slide to contact the outer race and three meters permit separation of the measurement into three bands, according to the settings of the sensitivity selection-switches above each meter. A loudspeaker produces a characteristic tone, deviations being detected readily.

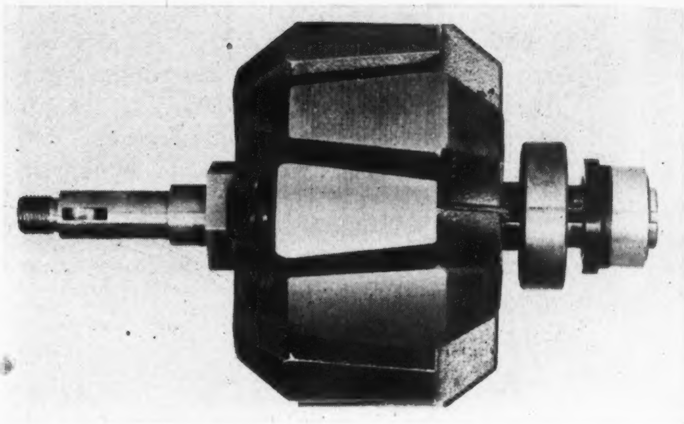
**Controlled-frequency** mounts on 1948 Buick engines employ a new vibration control system. As illustrated, below, the mounting is a three-point suspension, two at the front and one at the rear. It is designed so that the frequency of the engine on its mounting does not get in tune with other vibrations whether they are induced by the engine or by road irregularities. Front mounts are located on opposite sides of the engine near the center of the engine fore and aft, and about midway between the top and bottom of the cylinder block. They support the weight of the engine and control its torsional characteristics. The rear mounting is in two parts, one to support the weight of the power plant at the rear and the other to take the thrust from the rear wheels.

Synthetic rubber of the GR-S type is used for the mountings because of its internal frictional characteristics, giving it unique damping properties. In fact the synthetic is equal in performance to natural rubber mounts in combination with mechanical dampers. Previously, in dealing with engine mounts, designers usually stopped with a solution to the frequency aspect of the problem and made little effort to introduce damping, although it had been developed to a high state of perfection in chassis suspensions.

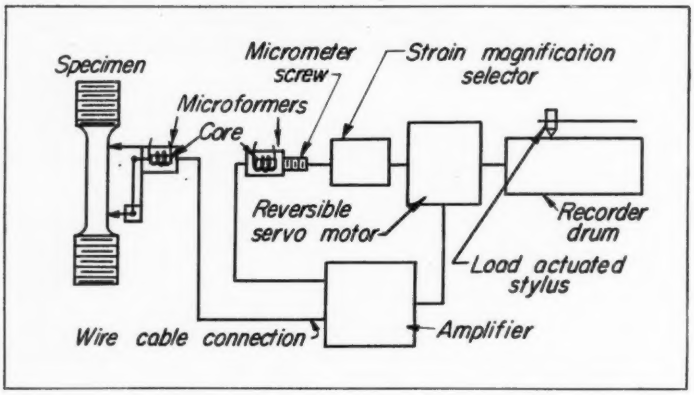



**Generating system**— consisting of an alternator, voltage regulator and rectifier—weighs somewhat less than conventional direct-current equipment of comparable output. The alternator, rotor and





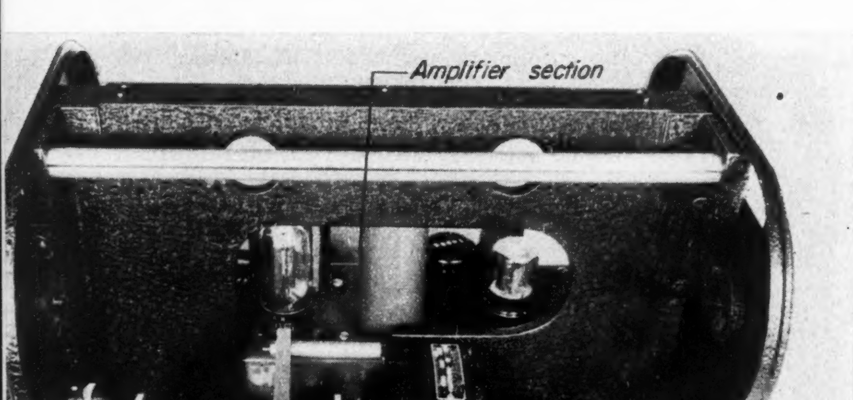
stator shown above, is of simplified design, having no commutator or rotating armature windings. Developed originally by The Leece-Neville Co. for police cruisers, the system generates sufficient amperage to keep batteries fully charged even though auxiliary equipment is used during long periods of engine idling time. On a six-volt system, 25 to 35 amperes are delivered at idle speed and 60 amperes up to top speed.




**Variable transformers** of the miniature type are utilized to control a new stress-strain recording instrument designed by The Baldwin Locomotive Works. The recorder for this instrument is shown at left with the drum and stylus removed to show the amplifier section. A schematic diagram for a tensile-test setup is also illustrated at left.

Change in deformation of the specimen actuates the movable core in the transformer coupled to the extensometer. This varies its output and unbalances an electrical circuit that inculdes a similar variable transformer in the recorder. The resulting electrical impulse is amplified to drive a servomotor which moves the core of a second transformer to rebalance the electrical circuit. The recorder drum is tied into the servo and thus moves in proportion to specimen deformation.

Corresponding values of applied load are recorded by movement of the stylus parallel to the axis of the recording drum. The stylus is actuated mechanically by the testing machine load indicator. Direction of rotation of the drum changes automatically with change in direction of deformation in the specimen. This system is useful for almost any pair of variables.



# Mechanism Proportions

## . . . solved exactly by application of simple geometry

By A. S. Hall
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SIMPLICITY and versatility of the quadric-crank, or four-bar linkage, makes it one of the most widely employed of all mechanisms. It appears in a multitude of machines and mechanical devices, either as the main mechanism or in series and parallel combination with others. Frequently the kinematic requirements of a design for a particular application are not stringent and the designer's problem is relatively simple, involving a selection of one design from a large group obviously available. In other cases, while the requirements are not too stringent, the possible solutions are not immediately apparent and the problem is slightly more difficult. A cut-and-try procedure, although frequently employed to arrive at a successful design, is likely to be wasteful of time and has the disadvantage of not showing all possible solutions, hence does not lead positively to the best design. This article will explain the application of plane geometry to exact solutions of a few of the quadric-crank design problems.

### Quick-Return Action Obtained

In *Fig.* 1 is shown a quadric-crank of such proportions that the driving crank, 2, may make a complete revolution while the lever, 4, oscillates through an angle $\theta$ less than 180 degrees. While lever 4 oscillates in one direction the crank turns through the angle $(180+\alpha)$; during the return oscillation of the lever the crank turns through the remaining angle $(180-\alpha)$. Thus a "quick-return" action is obtained if the crank turns at constant speed.

In the design of such a mechanism one or more of the link lengths, $L_1$ (distance $AB$), $L_2$, $L_3$, and $L_4$ and one or more of the angles $\alpha$, $\beta$, $\theta$, and $\phi$ may be fixed by the function which the mechanism is to perform. The problem is to determine the unknown dimensions. There are many conceivable design problems, depending upon which of the foregoing quantities are specified. Some of these problems have immediate and obvious solutions, others require more study. A few will be considered here to illustrate the usefulness of certain theorems and techniques of plane geometry.

LEVER RADIUS, $L_4$, AND EXTREME POSITIONS $\theta$ AND $\phi$ SPECIFIED: In this case a line *a-b* (*Fig.* 2) may be drawn and point *B* selected. The isosceles triangle

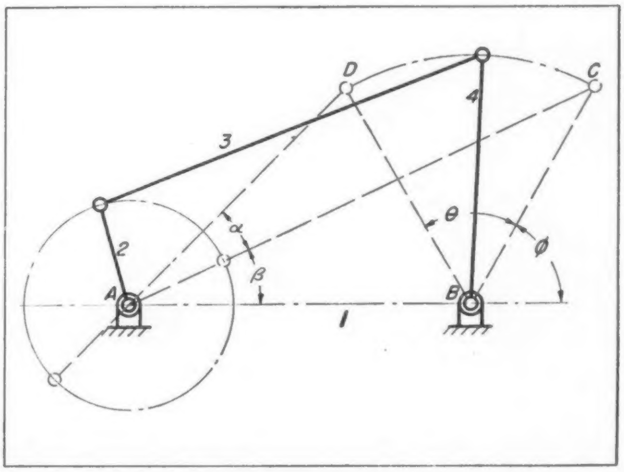
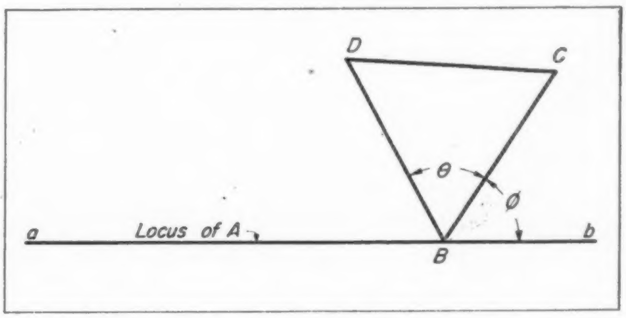



*Fig. 1—Above—Quadric crank, or four-bar linkage, with rotating crank and oscillating lever*

*Fig. 2—Below—Layout of specified conditions where length and extreme positions of lever are known*



*BCD* may then be constructed. Point *A* may lie anywhere on line *a-b* to the left of *B*. This line is the locus of possible positions of *A* for the specified conditions. In order to satisfy a fourth condition point *A* must have a particular location on this line. A good technique is to neglect the condition that $\phi$ must be a fixed angle and determine a second locus for *A* by applying the fourth condition to be satisfied.

*Problem* 1 (*Fig.* 3): *Center distance,* $L_1$ *specified in addition to* $L_4$, $\theta$ *and* $\phi$. Neglecting the condition that $\phi$ must be a fixed angle and applying the condition that $L_1$ must be a fixed length determines for point *A* a second locus which is an arc of the circle having *B* as center and $L_1$ as radius. The intersection of this arc with line *a-b* determines the location of point *A*. The unknown dimensions, $L_2$ and $L_3$, are then determined from the relations

$$AC + AD = 2L_3$$

and

$$AC - AD = 2L_2$$

*Problem* 2 (*Fig.* 4): *Angle* $\beta$ *specified in addition to* $L_4$, $\theta$ *and* $\phi$. In this case point *A* must obviously

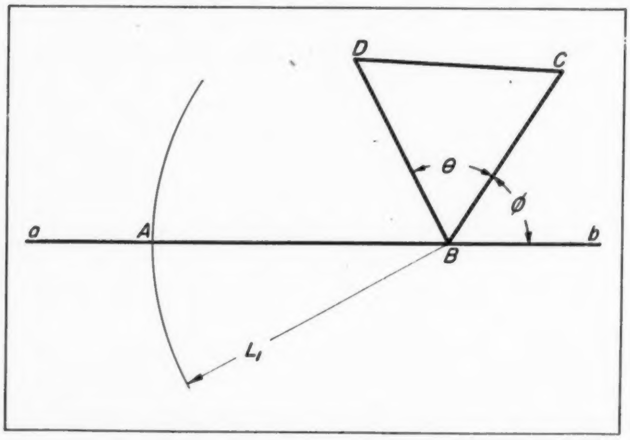
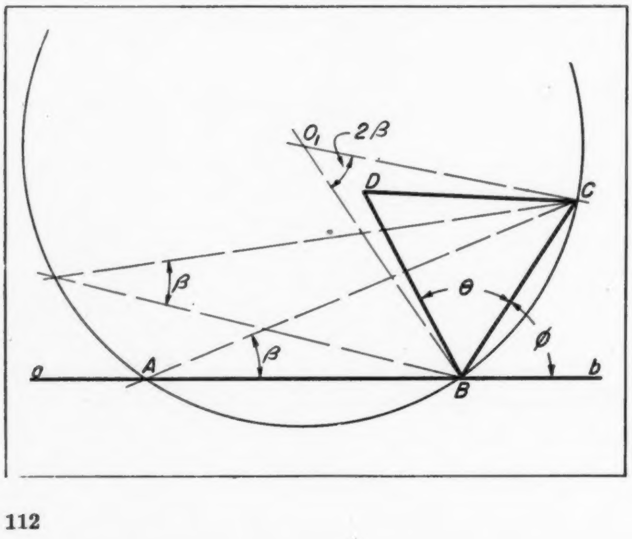



Fig. 3—Above—Construction for locating center line of crankshaft when center distance is specified

Fig. 4—Below—How to locate crankshaft center line when inner dead center position is specified




be located at the intersection of line *a-b* and a line through *C* making the specified angle $\beta$ with *a-b*. However, neglecting the condition on *A* determined by $\phi$ and applying the conditon determined by $\beta$, second locus of *A* is found to be the curve determined by the intersections of pairs of lines drawn through *B* and *C* at the angle $\beta$ with each other. That this locus is a circle arc becomes apparent when it is recalled that triangles with a common base and inscribed in a

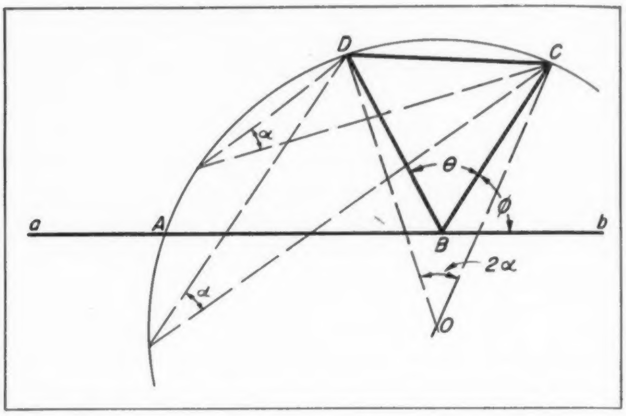



Fig. 5—Method of locating crankshaft center line when angle $\alpha$ is specified

circle have equal vertex angles (supplementary if vertices are on opposite sides of the base). In this problem the common base is *BC* and the vertex angle $\beta$. The circle center, $O_1$, is located at the vertex of the isosceles triangle having vertex angle $2\beta$ and constructed on *BC* as base.

*Problem* 3 (*Fig.* 5): *Angle* $\alpha$ *specified in addition to* $L_4$, $\theta$ *and* $\phi$. In this case the second locus for *A* is the curve determined by the intersections of pairs of lines through *D* and *C* at the specified angle $\alpha$ with each other. This locus is another circle with center at *O* as shown in *Fig.* 5.

*Problem* 4 (*Fig.* 6): *Connecting rod length* $L_3$ *specified in addition to* $L_4$, $\theta$ *and* $\phi$. For $L_3$ to be a fixed length the sum of the distances *AC* and *AD* must be a fixed quantity ($2L_3$), since $AC = L_3 + L_2$ and $AD = L_3 - L_2$. Hence the second locus for *A* in this problem is the curve determined by the intersections of pairs of arcs drawn with *D* and *C* as centers and radii whose sum is $2L_3$. This curve will be recognized as an ellipse with foci at *D* and *C*. Points of this ellipse are readily constructed. The intersection of the ellipse and line *a-b* locates point *A*.

*Problem* 5 (*Fig.* 7): *Crank radius* $L_2$ *specified in addition to* $L_4$, $\theta$ *and* $\phi$. In this case the second locus for *A* is determined by the intersections of arcs drawn with *D* and *C* as centers and having radii whose difference is $2L_2$. This is a hyperbola with foci at *D* and *C*.

The foregoing five problems cover the situations where lever radius and extreme positions are specified. The solutions of several other problems are immediately apparent. For example, if lever radius $L_4$, and stroke $\theta$, and the crank angles at dead center

positions $\beta$ and $\alpha$ are specified, the solution is found by locating point $A$ at the intersection of the circle locus of *Fig.* 4 and the circle locus of *Fig.* 5; if $L_4$, $\theta$, $\alpha$, and connecting rod length $L_3$ are specified point $A$ is located at the intersection of the circle of *Fig.* 5 and the ellipse of *Fig.* 6.

CENTER DISTANCE, $L_1$, AND ANGLES $\beta$ AND $\theta$ SPECIFIED: Line $AB$ can be drawn and points $A$ and $B$ located the specified distance $L_1$ apart, *Fig.* 8. Line $AX$ is drawn at the specified angle $\beta$ with line $AB$. Point $C$ may be chosen on line $AX$, with the restriction that angle $ABC$ must not be less than $\theta$ and the corresponding location of $D$ found by constructing isosceles triangle $BCD$. Two solutions are shown in *Fig.* 8, the solution $BC'D'$ for $\alpha = 0$ and the solution $BCD$ for $\alpha$ different from zero.

It can be shown that, as $C$ is allowed to move on its locus $AX$, $D$ will have for its locus the straight line through $D'$ at angle $\theta$ with line $AX$. The reasoning is as follows:

(a) Angle $BD'C$ = angle $BDC = 90 - (\theta/2)$ by construction. Hence the quadrilateral $BD'DC$ can be inscribed in a circle.

(b) Since $BD'DC$ is an inscribable quadrilateral the angles $CD'D$ and $CBD$ are equal. Angle $CBD = \theta$ by construction. Hence angle $CD'D = \theta$ and is a fixed angle.

Since $D'$ is a fixed point all possible positions of $D$

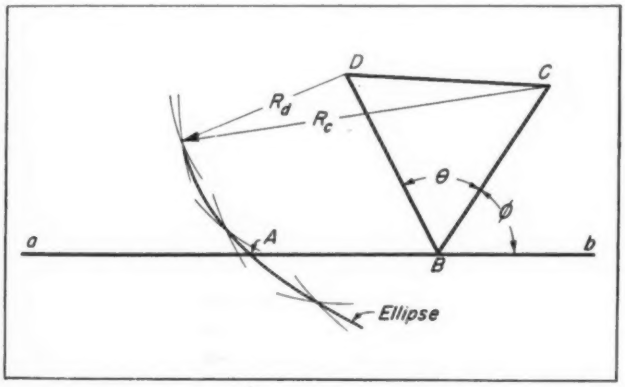
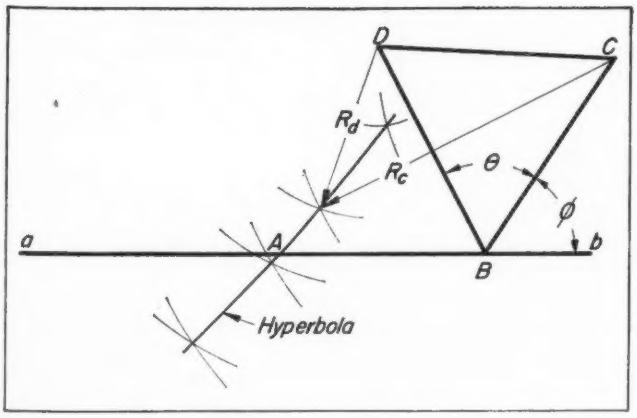



*Fig. 6—Above—Construction for crankshaft center line when connecting rod length is specified.* $R_c + R_d = 2L_3$

*Fig. 7—Below—Construction for center line of crankshaft when connecting rod length is specified.* $R_c = R_d = 2L_3$



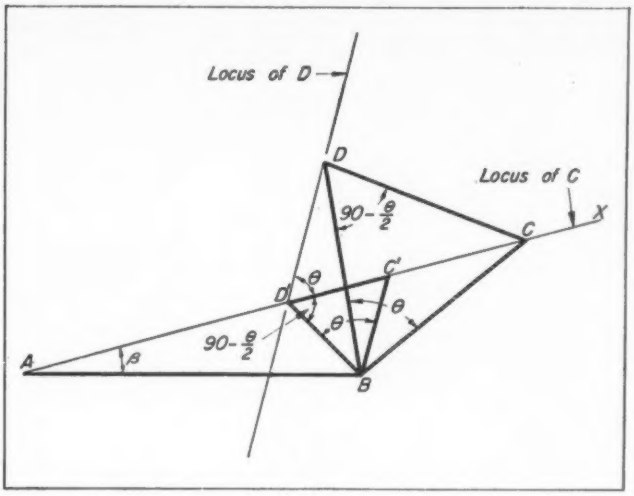
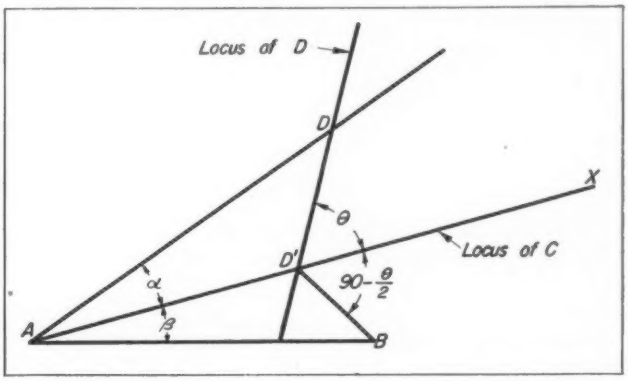




*Fig. 8—Above—Two possible solutions for lever arm length and extreme positions when $L_1$, $\beta$ and $\theta$ are specified*

*Fig. 9—Below—Determination of lever arm length BD when angle $\alpha$ is specified in addition to $L_1$, $\beta$ and $\theta$*




lie on the fixed line through $D'$ at the angle $\theta$ with line $AX$. It will be observed that $D$ cannot be chosen below line $AB$ as this would require lever 4 to pass a dead point in its oscillation.

*Problem* 6 (*Fig.* 9): *Angle $\alpha$ specified in addition to $L_1$, $\theta$ and $\beta$.* Neglecting the condition on $D$ determined by $\alpha$ it is possible to establish loci for points $D$ and $C$ as described in the foregoing. A second locus for $D$ is the straight line through $A$ at the specified angle $\alpha$ with line $AX$. $D$ is located at the intersection of its two loci and the location of $C$ follows directly.

Some special situations are of interest. If $\alpha$ is specified equal to $\theta$ the two loci for $D$ do not intersect, hence a solution is not determined. From another point of view the meaning is that the solution requires distances $AD$ and $BD$ to be infinitely great. However, the student of kinematics will recognize that this is a requirement which can be met mechanically by joining links 3 and 4 with a sliding joint rather than a pin joint. A solution is then the crank and vibrating arm mechanism shown in *Fig.* 10*a*.

An even more special situation is the case where $\alpha$ and $\theta$ are equal and $\beta = 90 - (\theta/2)$. In this case point $D'$ of *Fig.* 9 falls at $A$ and the two loci for point $D$ coincide. There are many solutions, one of which is illustrated in *Fig.* 11. Point $D$ may be chosen any-

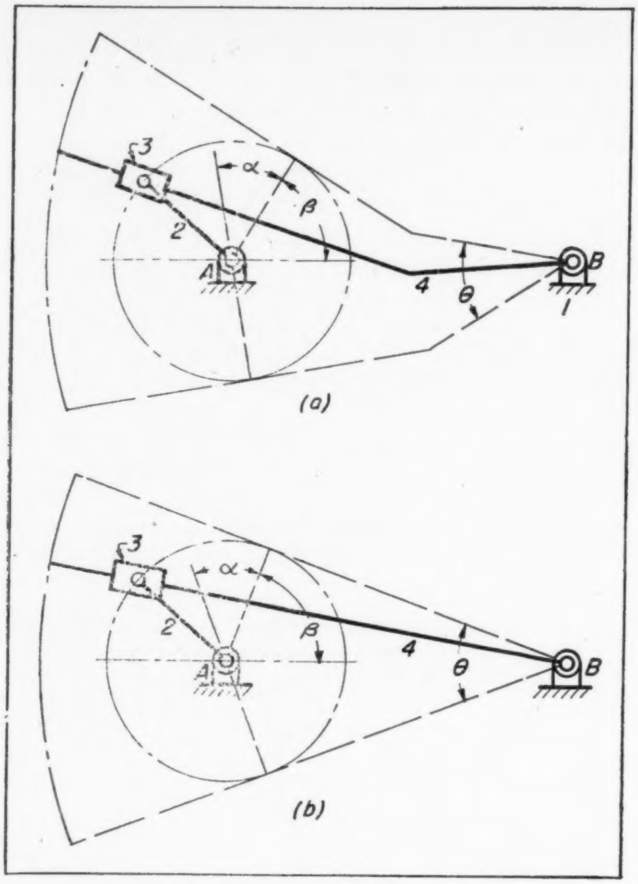



*Fig. 10—Crank and vibrating arm mechanisms, which are solutions for the special case $\alpha = \theta$*

where on its locus above line *AB*, including the infinitely distant point (the solution shown in *Fig.* 10*b*).

*Problem* 7 (*Fig.* 12): *Connecting rod length $L_3$ specified in addition to $L_1$, $\theta$ and $\beta$.* A second locus for point *D* can be established by neglecting the condition, *Fig.* 8, that angle *BDC* must be $90 - (\theta/2)$ and applying the condition that the sum of the distances *AC* and *AD* must be $2L_3$. The necessary construction work might be performed as follows, *Fig.* 12:

(a) On line *AX* lay out distance $AK = 2L_3$
(b) Choose several points, such as *C'*, on line *AX* and join to *B* with straight lines
(c) Draw lines *C'D'* at angle $90-(\theta/2)$ with lines *C'B*
(d) With *KC'* as radius strike arcs with *A* as center intersecting lines *C'D'*. These intersections determine points on a second locus for *D*. The intersection of this curve and the previously determined straight line locus determines *D*.

Center Distance, $L_1$ and Angles $\theta$ and $\alpha$ Specified: One solution (for a particular value of $\beta$) was described under *Problem* 6 (*Fig.* 9). The objective now is to determine the locus of point *D* when $\beta$ is allowed to vary. The reasoning, referring to *Fig.* 13, is as follows:

The angle *AD'B* remains constant as $\beta$ varies, since $\theta$ is fixed and, by the construction, angle *AD'B* is $90 + (\theta/2)$. Hence the variable triangle *AD'B*, having a fixed base(*AB*) and a fixed vertex angle, is inscriptible in a circle. The circle center is at *O'*.

As *D'* moves on its circle locus, point *D* describes a locus such that triangles *AD'D* are all similar. This follows from the observation that the angles of the triangle *AD'D* are fixed by the specified $\alpha$ and $\theta$,

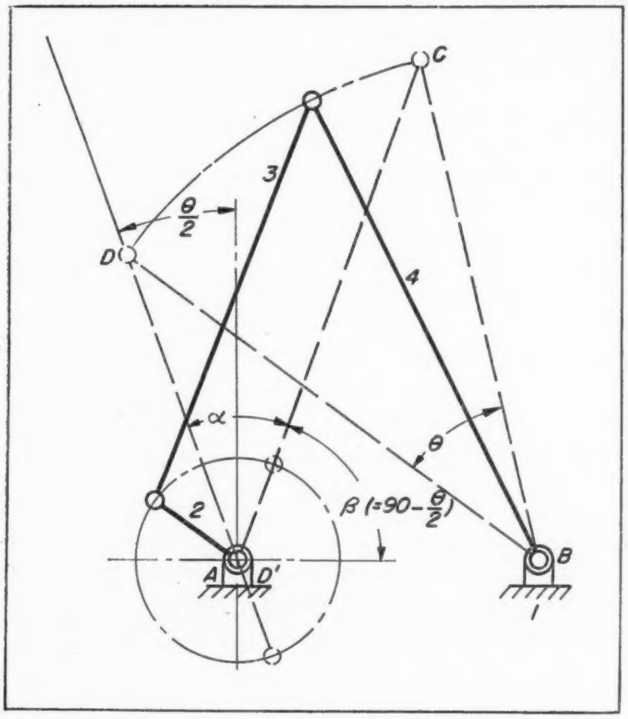



*Fig. 11—Above—A solution for the special case $\alpha = \theta$ and $\beta = 90 - \frac{1}{2}\theta$*

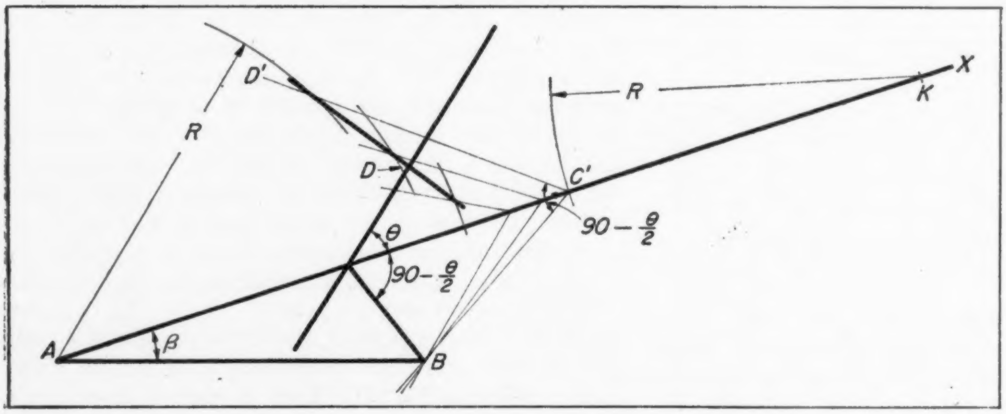



*Fig. 12—Left—Construction for point D when connecting rod length $L_3$ is specified. $AK = 2L_3$*

hence are not dependent on the varying $\beta$.

If, in a series of similar triangles having a common vertex (or a variable triangle having fixed angles and one fixed vertex), a second vertex lies on a circle then the third vertex will also lie on a circle.* Furthermore the fixed vertex and the centers of the two circles form a triangle similar to the given triangle. Hence, as $D'$ moves on the circle with center $O'$, $D$ will move on the circle with center $O$.

The necessary construction for determining the locus of $D$ is shown in *Fig.* 14. Two lines, drawn from $A$ and $B$ at the indicated angles with $AB$, locate center $O$ for the circle locus of $D$. The radius is length $OA$. Possible positions of $D$ are limited to that portion of the circle shown with a heavy line.

*Problem* 8 (*Fig.* 15): *Angle $\phi$ specified in addition to $L_1$, $\theta$ and $\alpha$.* $L_1$, $\theta$ and $\alpha$ determine the circle locus for $D$. The intersection of this with the line through $B$ at angle $\theta + \phi$ with $AB$ locates $D$.

In the special situation $\alpha = \theta/2$, center $O$ of the circle locus for $D$ falls at $B$. For all values of $\phi$ length $L_4$ will equal $L_1$, as illustrated in *Fig.* 16.

In all of these illustrative problem solutions the angle $\alpha$ has been taken counterclockwise from line $AC$. The reader may find it of interest to work out solutions of the same problems for $\alpha$ taken clockwise from line $AC$.

While the problems discussed here are few in number, and are limited to the four-bar linkage with rotating crank and oscillating lever, the general method of attack illustrated should prove of value in a much wider field. The design of mechanisms is, in large part, a geometric problem. The greater the designer's knowledge of the theorems and techniques of geometry, the easier will be his task.

* See "College Geometry" by Altshiller-Court, Johnson Publishing Co., article 54.

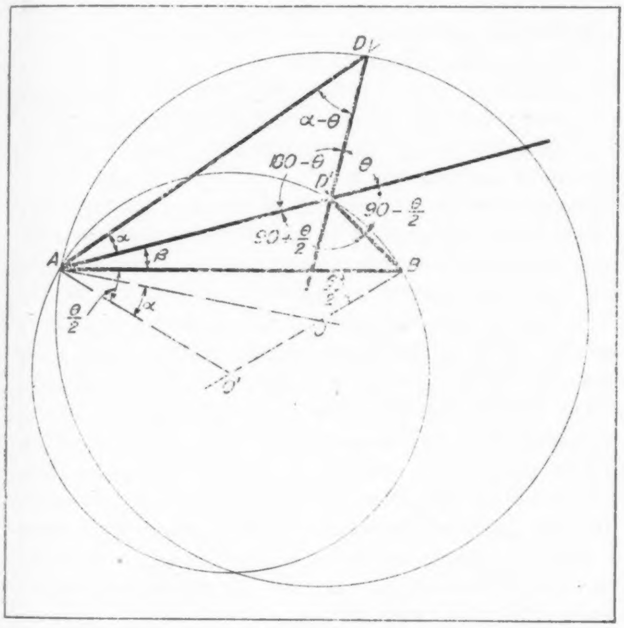
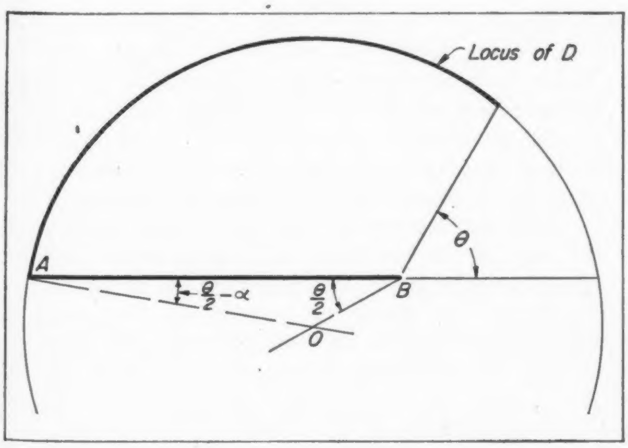



*Fig. 13—Above—Conditions establishing the locus of D when $L_1$, θ and α are specified*

*Fig. 14—Below—Construction for locus of D when $L_1$, θ and α are specified*



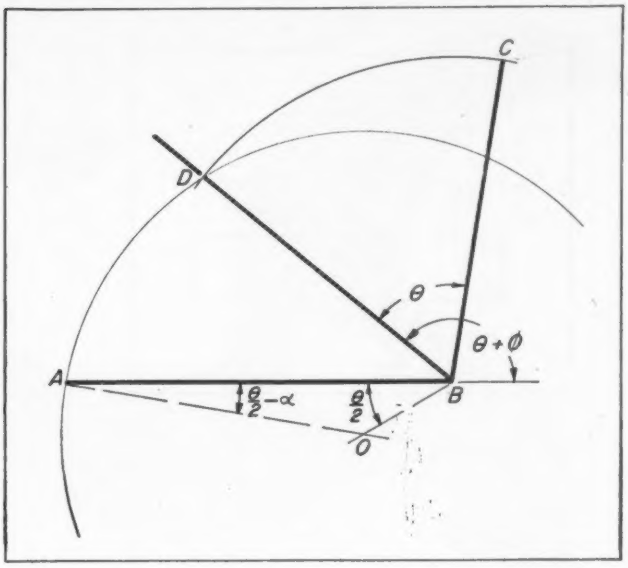
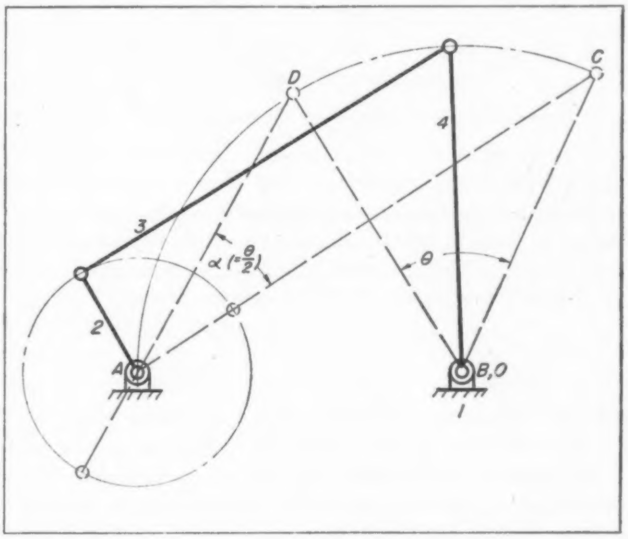




*Fig. 15—Above—Construction for locating point D when angle θ is specified*

*Fig. 16—Below—A solution for D for the special case α=θ/2. Length of lever equals center distance AB*



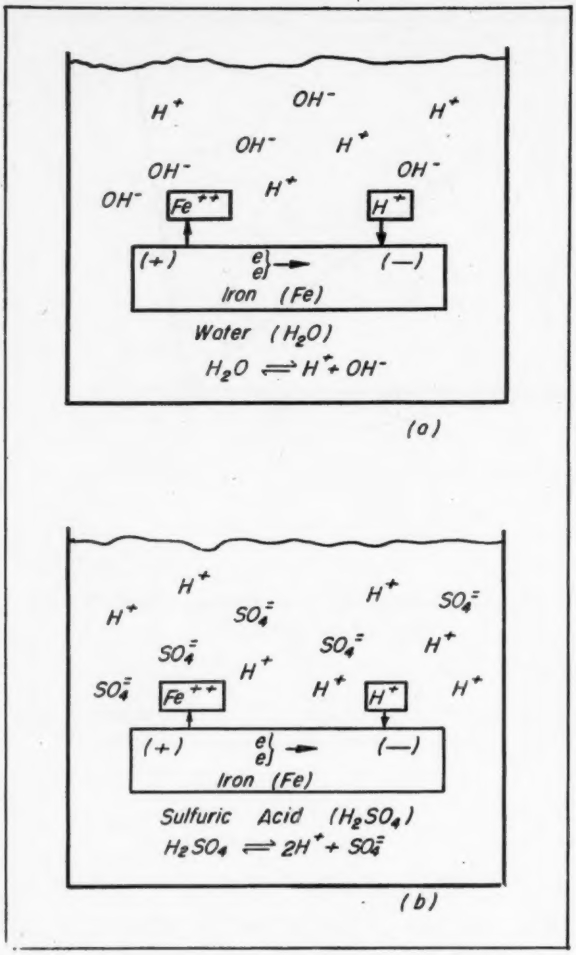



*Fig. 1—Direct-reaction electrochemical corrosion. (a) Rusting of iron in water wherein iron ions dissolve in the water. (b) Corrosion of iron in sulfuric acid formation of iron sulfate at anodic areas*

# How To Combat Co...


By E. T. Collinsworth, Jr.

*Chemical Section*
*Worthington Pump and Machinery Corp.*
*Harrison, N. J.*


IT IS estimated that approximately forty per cent of all iron and steel fabricated is used to replace machinery, equipment and structures which have corroded into uselessness[1]. It is also estimated that in this country alone, about $5000 worth of equipment is destroyed by corrosion every minute. Furthermore, replacement of equipment is not the only cost incurred by corrosion. There are other items, such as the following:

a. Disruption of production schedules due to shut-down
b. Inspection and maintenance costs
c. Contamination of products
d. Loss of efficiency in heat transfer equipment
e. Law suits due to equipment failures or personal injuries

In view of these facts, it becomes apparent that corrosion is a matter of considerable consequence in the design of machines subject to such attack. Unfortunately, to some machine designers corrosion is a vague and highly specialized subject understandable only to highly trained specialists. However, the acquirement of a practical, working knowledge of fundamental principles and concepts will serve admirably as an effective tool for combating corrosion in the design of machines.

To the machine designer the term corrosion indicates an action primarily involving metals, and its definition may be specifically stated as "the destruction of metals due to chemical and electrochemical action brought about by the external conditions existing in the metal's environment". For present purposes it will be assumed that all corrosive action occurs as one of two general types, namely, "direct chemical corrosion" or "electrochemical corrosion". The former does not involve the presence of water; the latter necessitates moisture in the environment. In either case, the accepted laws of thermodynamics and electrochemistry will dictate the course of reaction.

**Direct Chemical Corrosion:** This may be typified by the result of subjecting iron to an elevated temperature. When thus exposed, the iron reacts directly with the oxygen of the air to form complex oxides, commonly known as "mill scale". Other metals react in the same manner, and when exposed to certain other environments may again react to form still other compounds.

Typical examples are indicated by the following simple reactions:

$$2\,Zn + O_2 = 2\,ZnO$$
$$4\,Fe + 3O_2 = 2\,Fe_2O_3$$
$$4\,Al + 3O_2 = 2\,Al_2O_3$$

These show figuratively how zinc, iron and aluminum combine with oxygen to form their respective oxides. There are two facts to note about these examples: (1) The product is the result of the direct reacting elements, i.e., there is no displacement or substitution of one element by or for the other, and (2) there is no moisture present.

As moisture is present in nearly any environment for which a machine is constructed, it is easily seen that only a small per cent of the corrosion difficulties faced by the design engineer would be of the direct-chemical-corrosion type. For example, the design of the blading of a gas turbine would have nothing to do with the corrosion resistance except for the metallurgical consideration of the material.

**Electrochemical Corrosion:** This type of corrosion

[1] References are tabulated at end of article.

# rosion Through Design

## Part I—Causes of Corrosion

is the result of the displacement of one element in an aqueous phase by another element of the metallic state with an accompanying flow of electric current. Before this action can function, there are certain requirements which must be met: (1) A completed circuit for the current flow, (2) two electrodes and (3) a potential or driving force.

With electrochemical corrosion two reactions take place simultaneously upon the metal's surface at areas which may be immediately adjacent, but not identical. The anode (the corroded portion) has one reaction, constituting metal ions passing into solution, while the cathode has the other reaction, which involves hydrogen ions passing out of solution.

Existence of these so-called corrosion cells results from the dissimilarity of composition which exists not only in all metals but in their environments as well. These can be developed either from differences in

TABLE I

**Conditions Which Cause Corrosion**

| No. | Anode (corroded portion) | Cathode |
|---|---|---|
| 1 | Small grain size | Large grain size |
| 2 | Metal (oxygen—low concentration) | Metal (oxygen—high concentration) |
| 3 | Strained metal | Annealed metal |
| 4 | Base metal | Noble metal (galvanic corrosion) |
| 5 | Metal | Metal oxide on surface |

metal composition (TABLE I, Item 1) and/or environmental component concentration (TABLE I, Item 2) and/or physical condition of metal surface (TABLE I, Item 3). Other examples also are given in TABLE I. The following may be considered as types of electrochemical corrosion:

a. Direct reaction
b. Galvanic (two-metal) cells
c. Concentration cells (1) metal-ion cells, (2) oxygen cells

As can be deduced from TABLE I, it is a rare case where corrosion can be traced to any particular one of the types described in the foregoing. Consequently, it will be well to examine each, so that their combined effects may be understood for application to specific cases.

*Fig. 2—Direct acid attack on this pump impeller severely etched the surface, showing typical grain orientation*

**Direct Reaction:** In this type of corrosion, it is considered that the two electrodes are separated by an infinitesimal distance, an example being iron rusting in water. All metals have a tendency for an infinite number of plus and minus poles to exist upon their surfaces because of the inhomogeneity of their structures and compositions. Therefore, if they are immersed in an electrolyte which will complete the circuit, they will react as simple electrolytic cells. There are other variables influencing this tendency, called "rate factors", and these will be discussed later.

Consider the common phenomena of rusting. If a strip of iron (Fe) is totally immersed in water ($H_2O$) as illustrated by *Fig. 1a,* immediately there will exist on its surface innumerable plus (anodes) and minus (cathode) areas. For the sake of simplicity only two of these electrodes will be examined, being marked plus (+) and minus (−), respectively. At the anodes (+) of this simple cell, the iron will dissolve in the water in the form of iron ions ($Fe^{++}$) each having two plus charges. (An ion is considered to be an electrically charged atom or group of atoms.) With each ion going into solution, two electrons will remain with the parent metal. These migrate to the cathode (−).

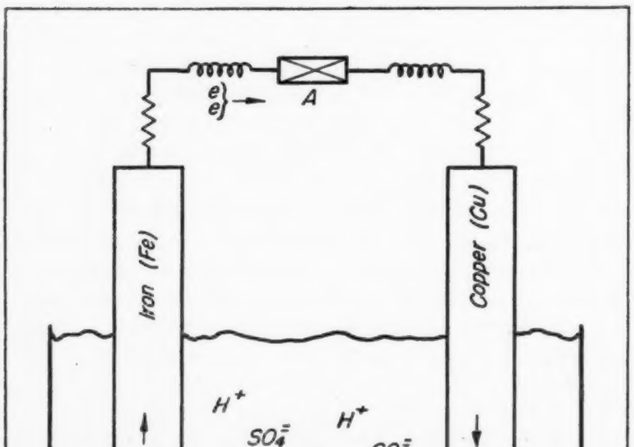
It is known that any solution contains, due to the result of disassociation (the breaking up of a compound into its component parts), the ions of the constituents of that solution. Thus, water is composed of hydrogen ions ($H^+$) and hydroxyl ions ($OH^-$) as indicated by the disassociation formula: $H_2O = H^+ + OH^-$: As shown by *Fig. 1a,* the hydrogen ion ($H^+$) reacts at the cathode with the free electrons to form hydrogen, $H^+ + e = H$. At the anodes (cor-

roding portion) the iron ions ($Fe^{++}$) react with the hydroxyl ions ($OH^-$) to form ferric hydroxide $Fe(OH)_3$ by a progressive reaction as follows: $Fe^{++} + 3\ OH^- = Fe(OH)_2 + (OH) = Fe(OH)_3$. This precipitate of ferric hydroxide is the commonly observed rust. Complete understanding of this simple corrosion cell is a basic step toward an understanding of electrochemical corrosion because all of the other types are specialized conditions of it.

Another illustration of the direct-reaction type of electrochemical corrosion is that of a metal with an acid, as shown by *Fig.* 1*b*. Here the sulfuric acid ($H_2SO_4$) is disassociated into hydrogen ions ($H^+$) and sulfate ions ($SO_4^=$). The reaction at the cathode would still be $H^+ + e = H$, but at the anodes, iron sulfate would be formed by $Fe^{++} + SO_4 = FeSO_4$. Corrosion of the metal-acid type is common, and it is to be emphasized that the difference between the electrodes in these discussed cases is actually infinitesimal and the attack has appeared overall. An example of a direct acid attack upon a machine part is typified by the pump impeller of *Fig.* 2 in which case the rate of corrosion was high enough to severely etch the surface so the grains appear clearly in their typical orientation.

**Galvanic Cells:** This is a term familiar to most no matter how slight their knowledge of corrosion. It can rightfully be called the "whipping boy" for the average corrosion investigator. The word "galvanic" itself, inferring its close relationship with electrochemical corrosion, may have something to do with the ease of manner with which corrosion problems are diagnosed in this category. Many designers have been informed that to place two metals which are widely separated in the "galvanic series" in contact invites all sorts of trouble because of "galvanic corrosion".

*Fig. 3—Example of galvanic cell is provided by electrically connected strips of iron and copper in sulfuric acid*



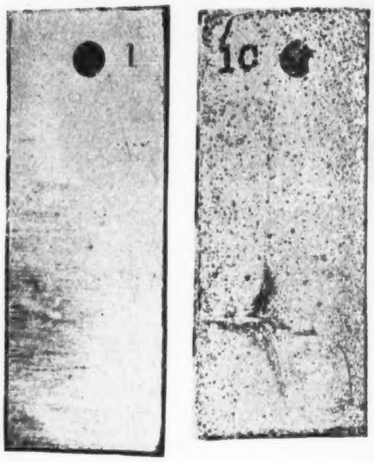



*Fig. 4 — Stainless steel specimens after exposure in acid sump of lead-lined spray pickling machine. Left—insulated from lead; right —galvanically connected to lead*

Actually galvanic corrosion is not nearly as important as is inferred. By far the majority of corrosion diagnosticians blaming "galvanic action" are misinterpreting the evidence. It is one of the easier types to avoid, especially in the original design of equipment. Only a few simple rules need be followed, which are a matter of course, if the phenomena is clearly understood.

## What Galvanic Corrosion Is

Galvanic corrosion may be defined as an accelerated electrochemical action due to two different metals being in electrical contact and exposed to the same electrolyte. It is well to note that concentration cells in special cases produce the same effect but since they do not fall within the terms of the above definition, they will not be discussed at this time.

With this type of corrosion the electrodes are separated by a measurable distance and exist as two different elements. However, the principle as described under "direct reaction" with the simple water-iron cell is exactly the same. Here, also, the behavior of this bimetallic combination depends upon certain "rate factors" with an electric current again being generated in the process. With this type, however, instead of infinitesimal areas for electrodes, there exists actual anodes and cathodes of different metals. This makes it possible to prove by measurements that the magnitude of this current is directly related to the amount of excess corrosion of the less noble metal beyond its normal corrosion rate under the existing conditions.

The "galvanic series", shown in TABLE II[2] indicates the tendencies of metals and alloys to form galvanic cells and to predict the probable direction of the galvanic effect, i.e., any metal in the series will corrode preferentially to any metal below it. It should be emphasized that this is an indication only, as the relationship is dependent upon environmental conditions. This holds true especially for those metals which demonstrate corrosion-resistant properties because of pro-

tective films or coatings acting as shields against corroding agents. These include the stainless steels, bronzes, aluminum, lead, and others. The action of such films will be discussed more thoroughly under "protective films".

An example of a galvanic cell could be a piece of iron electrically connected to a piece of copper, and immersed in a solution of sulfuric acid, as illustrated in *Fig.* 3. If switch A were to disconnect the circuit, both metals would tend to go into solution by "direct

TABLE II

**Galvanic Series of Metals and Alloys**

**Corroded End (anodic, or least noble)**

Magnesium
Magnesium alloys
Zinc
Aluminum 2S
Cadmium
Aluminum 17ST
Steel or Iron
Cast iron
Chromium-iron (active)
Ni-Resist
18-8 Chromium-nickel-iron (active)
18-8-3 Chromium-nickel-molybdenum-iron (active)
Hastelloy "C"
Lead-tin solders
Lead
Tin
Nickel (active)
Inconel (active)
Hastelloy "A"
Hastelloy "B"
Brasses
Copper
Bronzes
Copper-nickel alloys
Monel
Silver solder
Nickel (passive)
Inconel (passive)
Chromium-iron (passive)
18-8 Chromium-nickel-iron (passive)
18-8-3 Chromium-nickel-molybdenum-iron (passive)
Silver
Graphite
Gold
Platinum

**Protected End (cathodic, or most noble)**

reaction" as described and illustrated by *Fig.* 1*a*. However, in closing the switch, the more powerful galvanic current produced, as indicated by *Fig.* 3, superimposes the weaker solution tendency of the copper. The copper electrode actually would be protected by it at the sacrifice of the increased corrosion rate of the iron (notice that copper is below iron in the galvanic series).

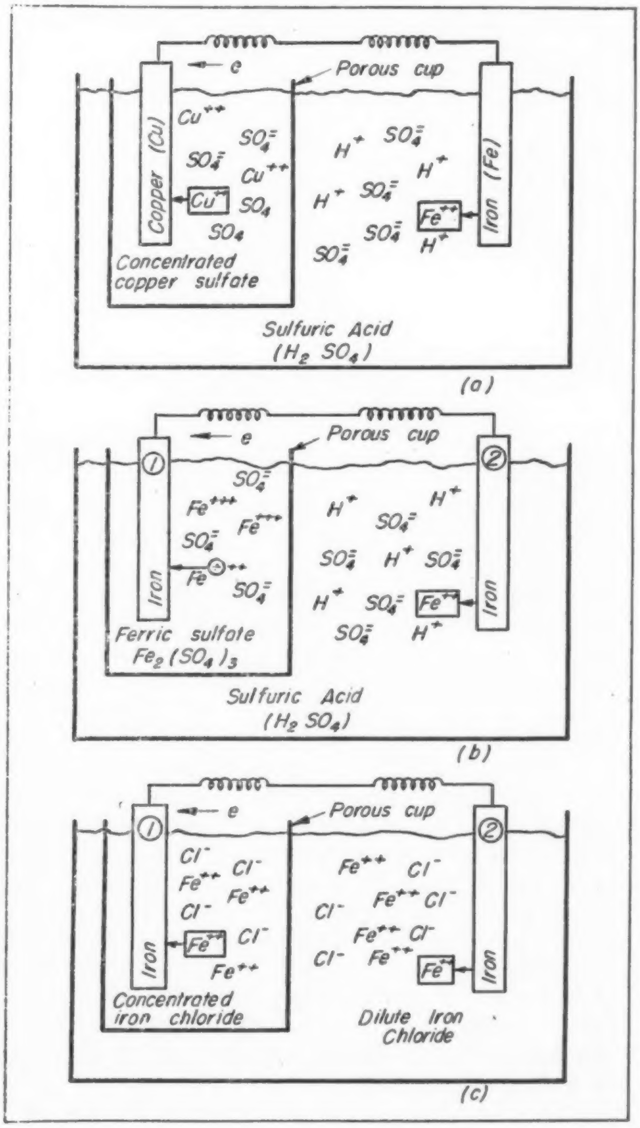
As previously stated, the galvanic series indicates only the probable direction of flow, as the relationship depends upon certain variables. These so-called "rate-factors" have been mentioned earlier in the discussion pertaining to "direct reaction", and they will be considered with concentration cells for a good reason. As these "rate factors" apply to any electrochemical corrosion action, they will be considered as a separate subject.

However, with galvanic cells there is one rate factor which is of utmost importance. This is the cathode-anode area ratio. It has no influence upon the metal potential difference, but it has a direct and important bearing on the current density, which is directly related to corrosion rate. A small anodic area galvanically connected to a relatively large cathodic area will result in rapid destruction of the anodic metal. If this area relationship was reversed, there would be little damage to the anodic metal, as the current would be distributed over a large area producing a low rate of corrosion.

An example of this would be the use of bronze fittings for iron pipe. Since the iron is anodic, galvanic corrosion can be expected to occur. But the anodic area is relatively so much larger than the cathodic bronze fittings, that the constant corrosion losses would be negligible. However, if bronze pipe was used with iron fittings, the life of these fittings might be only a few days.

An actual example of this type of corrosion is illustrated by *Fig.* 4. Both samples are of the same metal and were exposed to a 25 per cent solution of sulfuric acid at 170 F. Sample No. 1 was not subjected to galvanic corrosion as it was insulated from the lead in the system. It, of course, showed no weight loss after test. Sample No. 1*c* was attacked by this action

*Fig. 5—Metal-ion concentration cells. (a) Different electrolytes and different metals, (b) different electrolytes and same metal, and (c) different concentrations of same electrolyte, and same metal*



as it was connected and acted anodic to the large lead area, and showed high weight loss. Time of test was only 32 hours. (*Fig.* 8 also illustrates this type of corrosion.)

**Concentration Cells:** A "concentration cell" is that type of corrosion cell in which the electrodes may be considered as areas on the same or different metals either of which are subjected to portions of an electrolyte which differ in composition. For present purposes this type is divided into two divisions: (1) "metal-ion cells" and (2) "oxygen cells". This form of corrosion is an accepted, important one, the concept of which is becoming more and more important in diagnosing and preventing corrosion. To the design engineer, this might prove also to be the most difficult to understand.

METAL-ION CELLS: The metal-ion cell and the oxygen cell are similar. However, the former may be specifically developed by (1) different electrolytes and different metals, (2) different electrolytes and the same metal, (3) different concentrations of the same electrolyte and the same metal. The difference between these cells is the controlling factor for each type of reaction, being the metallic ion concentration and the oxygen concentration, respectively, in the electrolyte. The tendency for a metal to go into solution is related to the concentration of the ion of that metal in the contacting electrolyte. Therefore, the lower the ion concentration, the greater the tendency, and vice versa. It then can be reasoned that a current will flow in such a direction as to leave the metal at the point of low concentration, and enter at the point of high concentration. The above three developments for the metal-ion cell are shown by *Figs.* 5*a*, 5*b* and 5*c*, respectively.

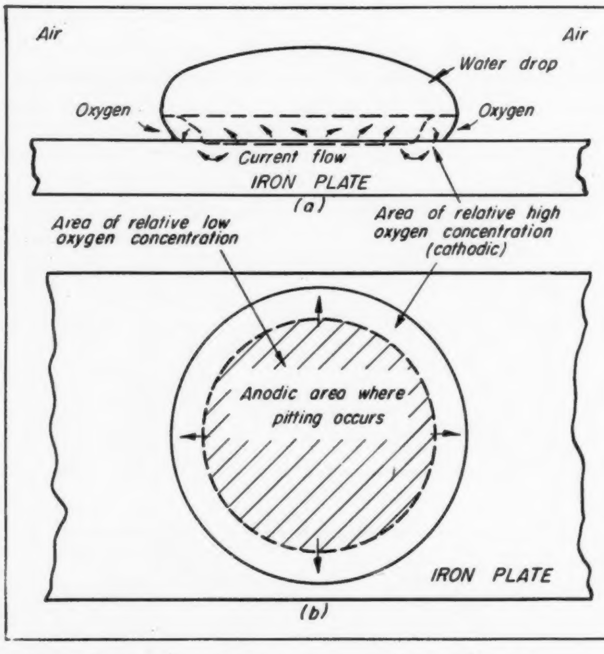



*Fig. 6—How drop of water causes pitting of iron*

With the arrangement shown in *Fig.* 5*a*, the solution of copper sulfate ($CuSO_4$) produces a high concentration of copper ions ($Cu^{++}$), thereby decreasing the tendency for the copper to go into solution. Consequently, the current flows from the copper (point of high concentration) to the iron (point of low concentration). The iron goes into solution forming ferrous sulfate ($FeSO_4$), leaving a minus charge on the parent metal. As is to be expected, upon receipt of these plus ions by the electrolyte, the electrochemical neutrality of the solution becomes upset. To restore this equilibrium, an equal number of positive charges must be displaced. As the porous cup is a conductor, the copper ions become reduced and deposit as metallic copper, completing the circuit. Due to this type of system, the connected iron is corroded at an accelerated rate, in excess of that which would result if it was attacked by the acid by "direct reaction" only.

In the second development, *Fig.* 5*b*, (different electrolytes and the same metal) approximately the same reaction results. There is a tendency for the ferric sulfate, $Fe_2(SO_4)_3$, to reduce to ferrous sulfate ($FeSO_4$), giving up a plus charge. This upsets the electrochemical equilibrium, causing the iron of electrode No. 2 to go into solution. Here, also, corrosion by electrode No. 2 is accelerated.

Different concentrations of the same electrolyte with electrodes of the same metal (third development) is illustrated in *Fig.* 5*c*. To reduce the concentration difference, there is a tendency for the iron to plate out on electrode No. 1. This, as previously described, causes the iron of electrode No. 2 to go into solution by accelerated attack.

It is to be noted that with this type of concentration-cell action, the same apparent results are produced as that of galvanic corrosion, i.e., accelerated attack. This information should be remembered in the design of machines the environments of which can be considered in this category. Practical application of this knowledge will be described in Part II of this series.

OXYGEN CONCENTRATION CELLS: Basically, as has been indicated, the metal-ion cell and the oxygen-concentration cell are similar. A metal or alloy may be exposed to electrolytes with variable oxygen concentrations. Here again the tendency is for the current produced by the concentration differential to leave the metal (corrode) at the point of low concentration. It would also enter the metal from the electrolyte at the point of high oxygen concentration.

## Common Phenomenon Cited

A good example of this action would be a drop of water on an iron plate. *Figs.* 6*a* and 6*b*, respectively, illustrate the sectional and top view of a drop of water. As would be expected, the iron would react with the water to form rust ($Fe(OH)_3$) as described by "direct reaction" (*Fig.* 1). This produces an oxygen depletion at the interphase; however, on the outside of the drop this is replenished by the diffusion of the atmospheric oxygen. It easily can be seen that an oxygen concentration differential can then exist

forming a concentration corrosion cell, as illustrated. Consequently, another action results other than that of direct reaction. A current is produced, leaving the metal at the anodic center of the drop, and re-entering it at the cathodic outside interphase. This produces a penetration of the metal at the center which has the appearance of a small pit. This example clearly illustrates a phenomenon which is extremely important to the machine designer inasmuch as it occurs in many machines.

It is to be understood that oxygen cells are not confined specifically to an electrolyte of water. This type also holds true with acids; with these, however, complex reactions sometimes are involved. The basic rules, as described in the foregoing, still govern the reaction. This behavior may be complicated by the passive effect of oxygen on the contacted metal, i.e., film effect. This will be discussed more thoroughly under "stainless steels". A metal-ion cell and an oxygen cell may sometimes oppose each other and nullify their respective effects.

PITTING: This form of corrosion usually is due to a combination of the electrochemical types described in the foregoing. As the name implies, it is the result of a highly localized corrosive action, causing abrupt penetrations in the metal. This localized action is due to a small anodic area having a high current density, which can be brought about by several factors including foreign inclusions in the metal (dirt and slag) as well as heterogeneity of the metal itself. This is an important consideration to the machine designer, as such localized attack is much more destructive to the strength of the machine member than is overall corrosion. An extreme example of this type of corrosion is illustrated by *Fig.* 7. This is a section of an impeller shaft of SAE 1020 steel from a deep-well turbine pump handling water contaminated with sulfur. *Fig.* 8, showing a nickel-chrome steel casing, is another example of pitting caused by galvanic action.

**Protective Films:** It has been stated that certain metals owe their corrosion-resistant properties to their ability to form protective surface coatings. These coatings are of varying thicknesses. Some, such as the coating on sulfated lead, can actually be scraped off. Others, such as the oxide film on stainless steels are only molecules thick. They are actually films of corrosion products which are formed when the metal is first exposed to its environment, inducing a self-stifling reaction, and producing a protective effect. Consequently, it can be said that metals falling within this category exhibit galvanic properties of the protective film rather than their true metallic characteristics. These metals include the bronzes, aluminum, lead, stainless steels and others.

**Stainless Steels:** Stainless steels are iron-base alloys formulated with sufficient chromium for their environments. They derive their corrosion resistance by virtue of a property called "passivity", which may be considered to be a state or condition of the alloy designated by its relative position in the "galvanic series". This is analogous to its more noble condition. If the state of "passivity" is lost, the alloy rises in the series, becoming less noble, and is considered to be in the "active state". This is clearly indicated by TABLE II.

True nature of the state of "passivity" has been the subject of many theoretical discussions for over a hundred years. However, the concept that it is due to an oxide film on the metal's surface should suffice for the average engineer. This film formation, only molecules thick, protects the metal from its corrosive environment. Examples of this are tableware, chrominum stripping on automobiles and a number of other such items.

It again should be emphasized that environmental conditions or "rate factors" are an important consideration in selecting the type of stainless steel for machine constructions. As the passive state is not necessarily a stable one, discretion should be used with the galvanic series.

*Fig. 7—Above—Extreme pitting of carbon steel shaft subjected to sulfur-contaminated water*

*Fig. 8—Below—Pitting of nickel-chrome steel casing after subjection to sulfuric acid in galvanic couple with lead*

Stainless steels are affected by all of the corrosion factors which have been discussed. However, as the protection is due to an oxide film, the presence of oxygen stabilizes the passive condition, enhancing its corrosion resistance properties. This action is contrary to that occurring on other metals not in this classification. As illustrated by *Fig.* 1, with iron, the more oxygen in the water, the higher the rate of corrosion. This is due to its depolarizing effect in keeping the cathodes (minus) clear of hydrogen by combining to form water ($H_2O$), and thus insuring free flow of the current. With a piece of stainless steel im-

mersed in the same water, the oxygen would enhance the outside film formation, giving it greater protection, and corrosion would not occur.

There are certain environments which are not suitable for stainless steel. If this is the case, the oxide film is destroyed and the steel acts with its true metallic properties as any other metal. The pump impeller of *Fig.* 2 is an example of a high-alloy stainless steel which was made active by improper application to hot hydrochloric acid service.

If the film is not completely destroyed and only localized breaks occur, a destructive pitting action will result. This is obvious, as the breaks exposing the base metal would act anodic to the relatively large cathodic stainless steel surface and the cathodic-anodic area ratio would come into effect. Pitting, due to chloride attack on stainless steels is a good example of this.

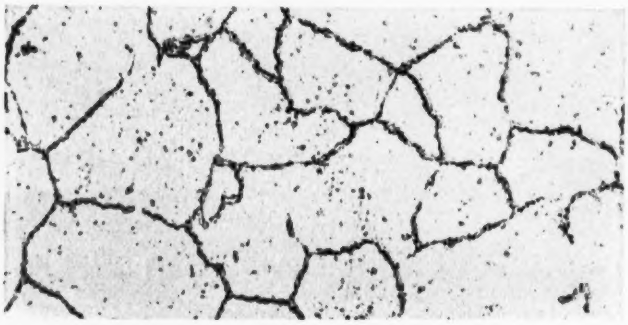
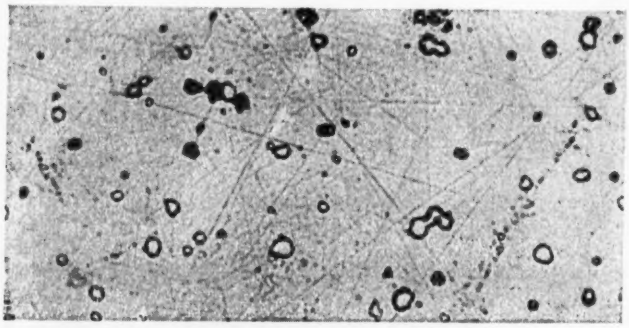


*Fig. 9—Above—In improperly heat-treated stainless steel, chromium carbides segregate along grain boundaries which are attacked preferentially*

*Fig. 10—Below—Correctly heat-treated stainless steel has no distinct segregation of carbides along grain boundaries; is less subject to attack*



There is another characteristic of stainless steel, knowledge of which is important to the machine designer. This is intergranular corrosion. With improper heat treatment (slow cooling) the carbon in the metal tends to combine with the chromium to form chromium carbides. These segregate along the grain boundaries of the metal, as illustrated by *Fig.* 9. The subsequent chromium depletion results in these grain boundaries being attacked preferentially (anodic) to the grain body. This obviously produces a weakened condition of the machine parts affected in this manner. The results are apparent and easy to diagnose inasmuch as the grains actually fall away from metal so attacked, with the surface usually taking on a gravelly appearance. The metallic sounding ring of the metal upon striking is also somewhat deadened.

This type of corrosion can be eliminated by using a stainless steel which has been correctly heat-treated with rapid-quench cooling to keep the carbon in solution without subsequent reheating. The use of stabilizers such as columbium added to the metal also helps to prevent this type of corrosion.

The stainless steel of *Fig.* 9, after correct heat treatment, is shown in *Fig.* 10. Here it is to be noted that there is no distinct segregation of carbides along the grain boundaries. This aspect is highly important with welding of stainless steels. In view of the above, it is apparent that if a proper technique is not used in welding the salutary effects of heat treatment will be destroyed and localized corrosion will result.

RATE FACTORS: The "rate factors" influencing all of these corrosion types may be noted and applied to their respective cases. Speller[5] lists these concisely as follows:

Factors associated mainly with the metal

1. Effective electrode potential of a metal in solution
2. Over-voltage of hydrogen on the metal
3. Chemical and physical homogeneity of the metal surface
4. Inherent ability to form an insoluble protective film.

Factors which vary mainly with environment

1. Hydrogen-ion concentration (pH) of the solution
2. Influence of oxygen in solution adjacent to the metal
3. Specific nature and concentration of other ions in solution
4. Rate of flow of solution in contact with the metal
5. Ability of environment to form a protective deposit on the metal
6. Temperature
7. Cyclic stress (corrosion fatigue)
8. Contact between dissimilar metals or other material as affecting localized corrosion.

These rate factors are not listed in the order of their relative importance. Individually, or in combination, they all exert a definite influence on the corrosion reactions and types which have been discussed. For those interested in a more detailed study of this subject, Reference 5 is highly recommended.

To the designer, theory is of negligible value without practical exemplified application. However, the full value of an example cannot be obtained if it cannot be expanded to include a combination of conditions. This would be impossible without an understanding of the fundamental concepts. With these, the reader is in a position to apply them to specific conditions to obtain the optimum results in the design of machines. This will be covered in detail in the second article of this series.

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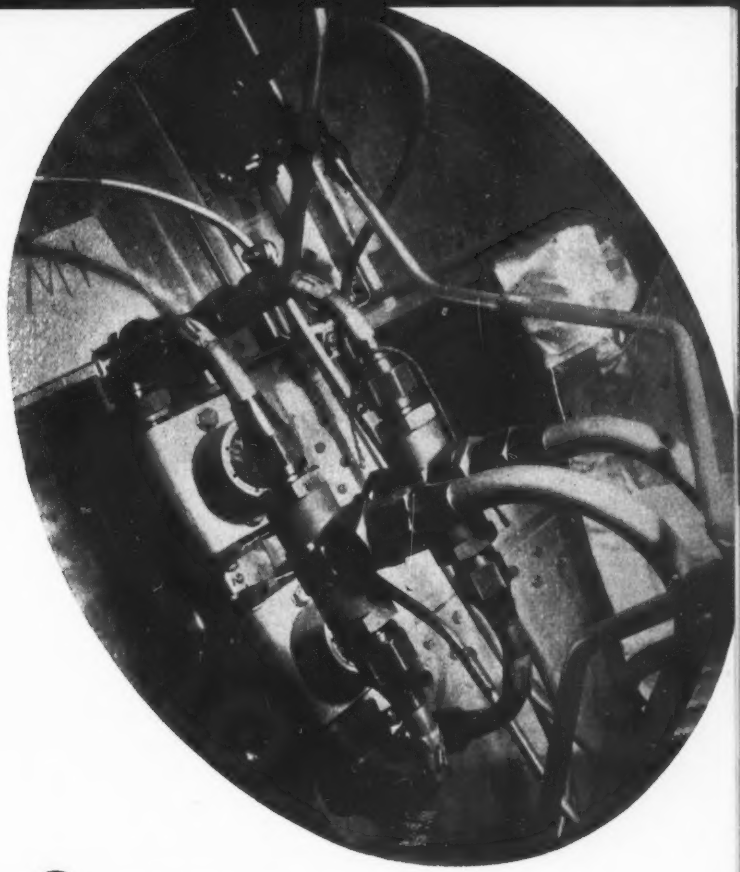

*3000-psi relief valves in the Martin 2-0-2 hydraulic system*

# Two-Pressure Aircraft Hydraulic System

***...combines benefits of weight saving, reliability, reduced maintenance, minimum fire hazard, and minimum external leakage***


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IN DEVELOPING the overall hydraulic system for the Martin 2-0-2, it was necessary to decide whether to use a 1500-psi or 3000-psi operating pressure. Weight being a matter of prime importance, it would appear, off-hand, that the 3000-psi system would be better. However, when reliability and other factors were considered, this was not entirely the case. For example, the windshield wipers were designed for 1500-psi operation and would have to be adapted for the higher pressure. Similiarly, the maximum brake pressure required is approximately 1100 psi, for which a 1500-psi system pressure is adequate.

In addition to reliability and weight, external leakage of hydraulic fluid is a factor which causes no end of annoyance and expense to operators. Related to design, this factor influences the choice of valves and the system pressure. Obviously, it is easier to seal a 1500-psi system against external leakage than it would be to seal a 3000-psi system. Correspondingly, it is easier to seal externally against the 300-psi return-line pressure with a slide valve than to seal either a 1500-psi or 3000-psi poppet type valve against external leakage at full line pressure. Of course, it is true that slide valves leak more internally than do poppet valves, but this can be allowed for in design and is of little or no concern to the operator as long as external leakage is minimized.

Choice of valve type, therefore, was another factor to be weighed before selecting the system pressure. Much satisfactory experience was available with 1500-psi poppet valves, but little could be said for the 3000-psi selection. To further complicate matters, four pumping systems are possible, all of which affect the choice of valve type and system pressure. First, there is the closed-center, variable-volume pump combination. With this, the entire system is at full pres-


This article is an abstract of a paper presented at the SAE National Aeronautic meeting in New York.

sure at all times and, at 3000 psi, external leakage and fire hazard would be quite a problem. The pump itself is another unknown and, therefore, a weak link. Service experience with variable-volume pumps is limited and at best is far inferior to the constant-displacement types.

Second, the closed-center, constant-volume pump, unloading-valve combination was considered. Here again, a considerable proportion of the lines are at full 3000-psi pressure at all times, making external leakage a problem. Then too, the unloading valve would work itself to death if internal-leakage slide valves were used.

## Combination System Required

Finally, the open-center system or a combination of systems is available. The obvious advantage of the open-center system is the low pressure to which lines and equipment are subjected at all times except during unit operation. The basic disadvantage is lack of stored pressure to operate essential units in the event of failure in the pumping system. Some combination of systems, therefore, seemed the only way to obtain maximum effectiveness for all the basic requirements. Using a 1500-psi, closed-center system for brakes, steering and windshield wipers and 3000-psi open-center, constant-volume pump system for landing gear and flaps was the combination desired.

Preliminary attempts to obtain this combination indicated the development of new equipment of questionable reliability. Because of this and the pressure of time schedules, work was started on the next best bet, which was a 3000-psi closed-center, variable-volume pump combination for all units. Previous experience with the construction and thorough testing of a complete airplane 3000-psi mock-up, plus the design of a 3000-psi system for an experimental Navy airplane, indicated that sufficient development work had been done on the basic problems to make this goal entirely feasible. The closed-center, variable-volume pump combination was favored over the closed-center, unloading-valve system to avoid the pressure surges usually encountered with unloading valve operations under conditions of high pressure and flow. It also permitted the use of simple slide valves, having a minimum liability of external leakage and a reasonable amount of internal leakage, without incurring the wear of frequent unloading valve "cut-in".

At this point in the design, Vickers, Inc., who had been working with us on our hydraulic equipment problems, submitted a proposal for a combination system similar to that which had previously been considered. The Vickers proposal consisted of a combination 1500-psi closed-center system and a 3000-psi open-center system, but also included a novel means of effecting this combination in the power circuit. The power circuit was set up in the Vickers laboratory and demonstrated to be simple and practicable. Such a basic system appeared to be the best compromise to attain the weight saving advantages of 3000 psi, and the reliability and other basic requirements.

Action of this system in normal flight is such as to maintain a continuous open-center flow through the 3000-psi circuit and at the same time to automatically maintain a margin of 1250 to 1500 psi in the closed-center circuit. This is accomplished by the combination of a modified Vickers 3000-lb unloading valve and a pressure reducing valve. Unloading valve modifications consist of reducing the unloading range of 1250-psi cut-in and 1500-psi cut-out and the use of a separate pilot pressure line from the accumulator. The latter serves to produce the cut-in and cut-out signals in lieu of the normal pressure feed - back through the accumulator port. The pressure reducing valve is mounted in the accumulator line adjacent to the unloading valve and is set to reduce from 3000 psi to 1700 psi. This unit also incorporates an integral relief valve.

A simplified diagram of the main pressure circuit is shown in *Fig.* 1. Landing gear and wing flaps are in the 3000-psi open-center circuit while nose-wheel steering, windshield wip-

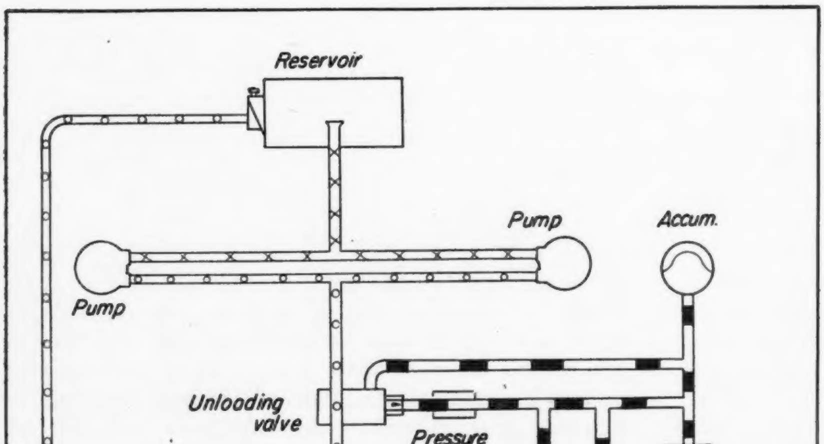



*Fig. 1—Normal flight condition of main pressure circuit. Closed-center system is charged while pump flow idles through open-center system*

ers and brakes are operated from the 1500-psi closed-center circuit. This diagram illustrates the normal flight condition with the closed-center system charged to 1250-1500 psi and with the entire pump flow idling through the open-center circuit. In this condition, the open-center system pressure is approximately only 300 psi as a result of normal pressure drop through lines and valves.

Conditions existing when the unloading valve is in the "cut-in" position and the pumps are charging the closed-center system are shown in *Fig.* 2. This occurs whenever the system pressure drops below 1250 psi due to valve operation or internal system leakage. As the accumulator fills and the pressure increases to 1500 psi, the unloading valve is actuated to the "cut-out" or open-center position. The "cut-in" and "cut-out" pressure signals are transmitted directly to the unloading valve from the accumulator through a separate line.

When the landing gear valve is actuated, the conditions shown in *Fig.* 3 exist. Open-center flow is cut off at the landing gear valve and the pump flow is directed to the landing gear cylinders. The system

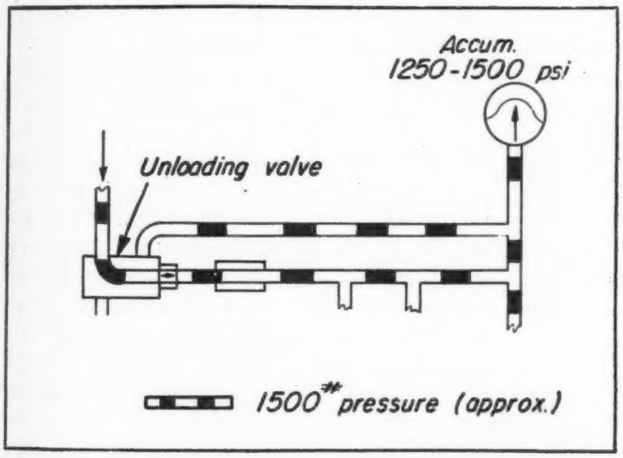



*Fig. 2—With unloading valve cut in, closed-center system is charged to about 1500 psi directly from pumps*

pressure then builds up to that required to move the landing gear and it is here that the pressure reducing valve comes into action. Assume that, at the time the landing gear valve is actuated, the closed-center system pressure is 1400 psi. As the system pressure builds up above 1400 psi due to landing gear loads, the check valve in the unloading valve is opened and the pump flow splits, part going to the landing gear cylinders and part to the closed-center accumulator. As the accumulator fills and the pressure reaches 1700 psi, the pressure reducing valve shuts off and holds the closed-center pressure at that value. The entire pump flow then goes to the landing gear cylinders to complete that operation, after which the landing gear valve automatically returns to the open-center position. The system is then back in the same condition illustrated by *Fig.* 1 except that the closed-center system pressure has been raised to 1700 psi. Wing flap operation results in a similar

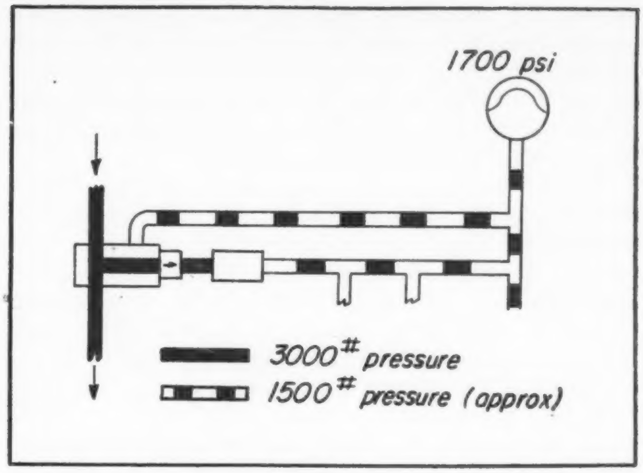



*Fig. 3—How pressure is built up in open-center system for landing gear or wing flap operation*

action in the pressure circuit.

An important functional feature of the system is the matter of priority of operation. The closed-center 1500-psi circuit has priority over the open-center circuit at all times due to the unloading valve and pressure-reducing valve action, which directs pump flow to this system at any time that the pressure drops below 1250 psi. This insures that windshield wipers will not be slowed up due to lowering flaps and landing gear during the approach. It likewise insures against loss or impairment of brakes, wipers, and steering while taxiing due to raising or lowering flaps. If the pressure required to operate flaps on the ground is less than 1250 psi and the closed-center system pressure drops to this figure, the unloading valve will momentarily cut in and stall the flaps until the closed-center pressure builds up to 1500 psi. At this point, the unloading valve will cut out and flap operation will be resumed. In flight, the pressures required for flap and landing gear operation will be above 1250 psi, thus automatically keeping the closed-center system fully charged and preventing any interruption of flap operation due to unloading valve cut-in. Continuity of operation of the landing gear cycle is assured in a similar manner.

Landing Gear Operation: With an open-center system, the landing gear handle must be latched in the neutral or mid-position at all times when the landing gear is not being actuated. In this position the pump flow passes straight through the valve and is blocked off from the cylinder ports. The cylinder ports are open to one another and the return port. This prevents any inadvertent build-up in pressure in the cylinders due to thermal expansion or leakage from the open-center pressure line.

To raise the landing gear, the handle is unlatched and raised to the up position. In this position the pump flow is directed to one side of the landing gear pistons, the return flow from the other side passing out through the valve return port. The landing gear handle must remain in the up position until the landing gear is up and locked at which time the valve must be returned to the open-center or neutral position. It was decided that this handle operation must

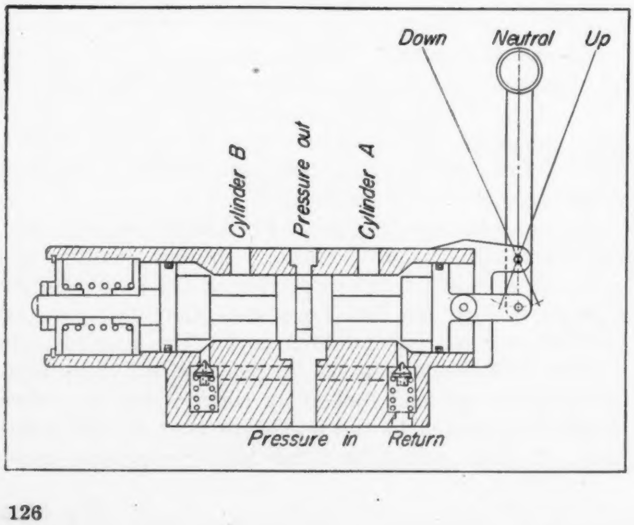
be entirely automatic once the pilot has moved the handle from the neutral to either the up or down position. This permits the pilot to concentrate on other duties while the landing gear is in motion. It also insures against his forgetting to return the handle to neutral after the operation is complete.

Several methods for accomplishing this automatic operation, which had previously been used, were considered but found to have undesirable features. Hydraulic pressure could readily be utilized to hold the valve in the operating position with a centering spring to provide the return to neutral. The main problem was to find a suitable means to signal to the valve when the landing gear operation was complete. Solution was found in utilization of the return flow from the cylinders. With this arrangement, return flow from the cylinders is utilized to hold the valve in the operating position against the action of a centering spring. When the return flow stops, signalling completion of the landing gear operation, the centering spring "takes over" and returns the valve to neutral.

In the schematic drawing of the valve, *Fig.* 4, the control is shown in the neutral position. In this position the pump flow enters the "pressure in" port, passes around the spool and out the "pressure out" port. It is sealed from connection with the cylinder ports by the spool lands on each side of the open-center ports. The two cylinder lines are vented to each other and the return line through the small holes in the two return poppets. For the automatic operation cycle, the pilot moves the valve handle to the right or up position and the spool moves to the left against the action of the centering spring, admitting flow from the "pressure in" port to cylinder port *A*. At the same time, the "pressure out" port is blocked and the opening to the right-hand return poppet is closed off. Pressure then builds up in the pump pressure line and starts raising the landing gear.

Return flow from the cylinders enters cylinder port *B* and lifts the left-hand return poppet off its seat, flowing across to the right-hand return chamber and out the return port. Since it takes about 80 psi to

*Fig. 4—Cross section through landing gear valve, showing centering spring and return poppets*




raise the return poppet against its spring, this amount of back pressure is built up in the left-hand side of the valve and cylinder line *B*. This pressure acts on the differential area of the valve spool holding it over in the full left position against the action of the centering spring. Thus, the pilot can release the handle and it will automatically stay in this position as long as return flow enters port *B*.

When all gears are in the up and locked position, the return flow to port *B* stops and the return poppet re-seats. The valve spool then returns to the neutral position by action of the centering spring. In moving to neutral, the valve spool displaces a small amount of fluid which passes to the return line through the small hole in the return poppet. The landing gear down cycle is accomplished in a similar manner by moving the control handle to the left. The valve is designed to provide this automatic operation at all pump flows from 16 gpm (take-off output) down to ¾ gpm (emergency electric pump output).

## Proper Operational Sequence Assured

Referring to the hydraulic system diagram, *Fig.* 5, it will be noted that the return flow from the landing gear cylinders goes into the main open-center return line from a separate return line on the valve; by-passing the flap valve. Thus, if wing-flap operation is attempted while the landing gear is in operation, no action takes place until the landing gear operation is complete and pump flow to the flap valve is resumed. In the instant that the landing gear up operation is complete, but before the valve has had a chance to return to the neutral position, the pump output flows over the twin main system relief valves at approximately 3000 psi. Since these relief valves constitute the normal pressure-regulating means for the 3000-psi system, they are installed in duplicate to provide an additional safety feature against excessive system pressure.

All landing gears on the 202 retract up and forward against the airstream. Thus, for down operation, it is only necessary to furnish power to release the up-locks and the gears will go down and lock by action of gravity and airload. Since it requires only approximately 500 psi to release the up-locks, the system pressure in the down direction has been limited to 900 psi. This is accomplished by installing dual relief valves in the landing gear down line and results in an appreciable weight saving in the landing gear, support structure and cylinders. It also saves a considerable amount of needless wear and tear on the pumps and system.

In case of complete failure of the hydraulic system, a simple cable-operated mechanical up-lock release system is provided with a pull handle accessible to the crew. The normal landing gear control lever may be left in the neutral position when lowering the gear by means of the emergency up-lock release.

Nose Wheel Steering: Power steering is accomplished by a steer wheel mounted on the console to the left of the pilot and nose wheel movement is proportional to steer wheel movement. The nose wheel is shimmy damped at all times and is automatically

free to caster when the pilot is not steering. Provisions are made to prevent steering when the nose wheel is not in contact with the ground with the landing gear either extended or retracted, thus making this part of the operation fool-proof.

In the cutaway diagram of the steer hydraulic system, *Fig.* 6, the steer shutoff valve is shown held in the open or steering position, indicating that the landing gear is down. In this position pressure from the 1500-psi closed-center system is free to pass straight through to the steer control valve. The steer control valve is shown in the neutral or spring-centered position. In this position pressure is blocked from both cylinder ports. It will be noted, however, that the two cylinder ports are open to one another and to the return port. This feature vents one side of the steer cylinder to the other, thus permitting the nose wheel to caster.

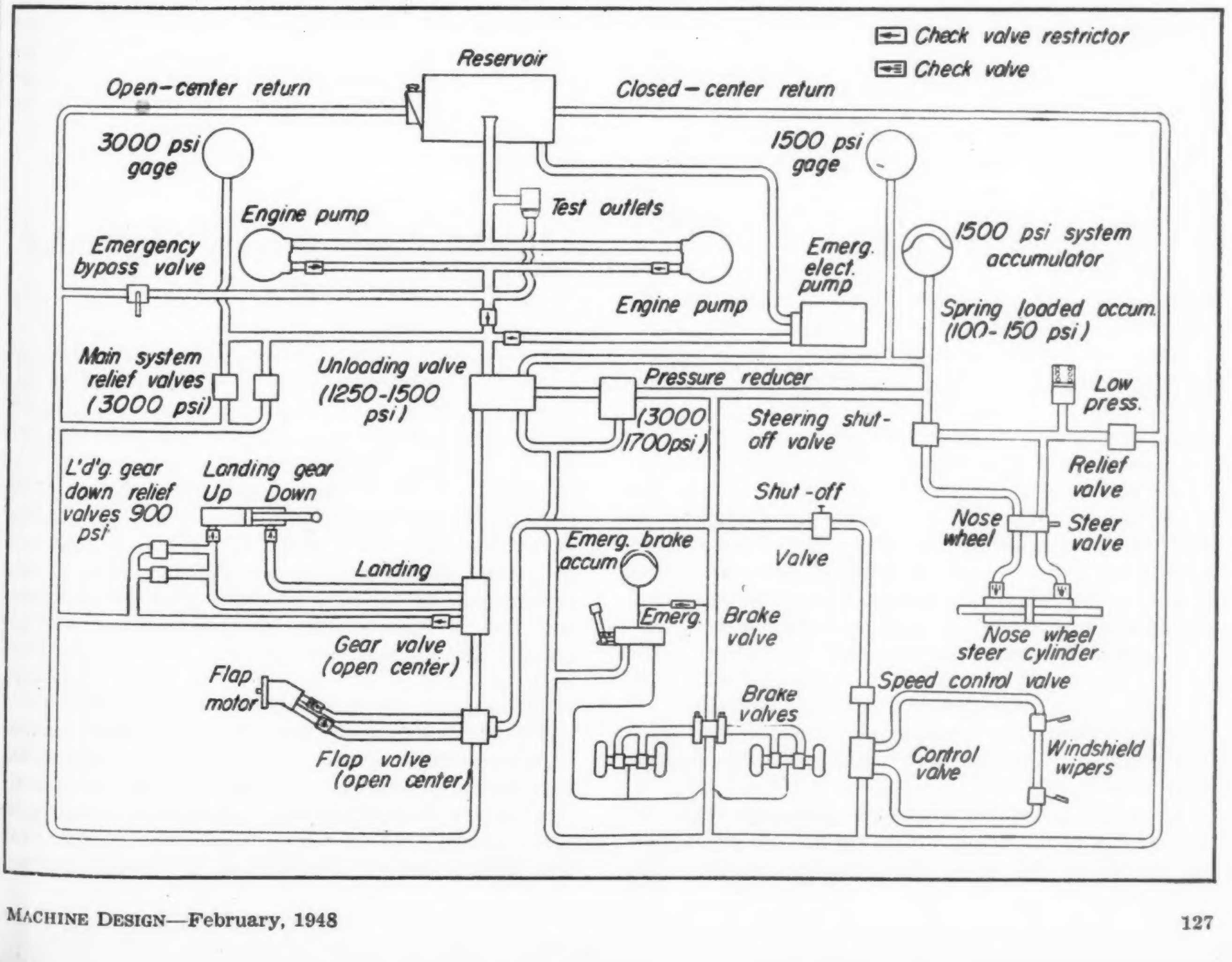
## Restrictors Damp Out Shimmy

Shimmy damping is accomplished primarily with the check valve restrictors mounted on each end of the steer cylinder. These valves permit free flow to the cylinder, but restrict the flow from the cylinder. As long as the cylinder is kept full of fluid, this obviously would provide shimmy damping, since the rate of movement of the steer piston in either direction is controlled by a restriction to flow.

Additional provisions are necessary, however, to insure that the steer cylinder is kept full of fluid at all times. These provisions consist of a low-pressure relief valve and a small spring-loaded accumulator installed in the return line from the steer valve. These units serve to keep the cylinder lines pressurized to approximately 100 psi at all times. When the wheel casters, thus forcing fluid from one side of the cylinder, the relief valve insures that the fluid passes through the steer valve and into the other side of the cylinder instead of flowing out the return line and causing a partial evacuation of the steer cylinder. The accumulator serves to maintain pressure even though some amount of external or internal leakage should occur.

In a typical steering cycle, assume the steer valve spool to be displaced to the right by movement of the pilot's control wheel. As the spool moves over against the action of the centering spring, the connection between the right cylinder line and return line is blocked off, followed by unporting of this cylinder line to the pressure port. Thus, 1500 psi pressure enters the right cylinder line, opens the check valve restrictor and acts on the steer piston causing it to move to the left and turn the nose wheel. Fluid in the left side of the steer cylinder is forced out through the orifice in the left check valve restrictor, through the steer valve to the return line. The low-pressure relief valve then blocks the flow until the spring-loaded accumulator is filled, at which point the relief valve cracks and the fluid flows into the system

*Fig. 5—Diagram of complete two-pressure hydraulic system*



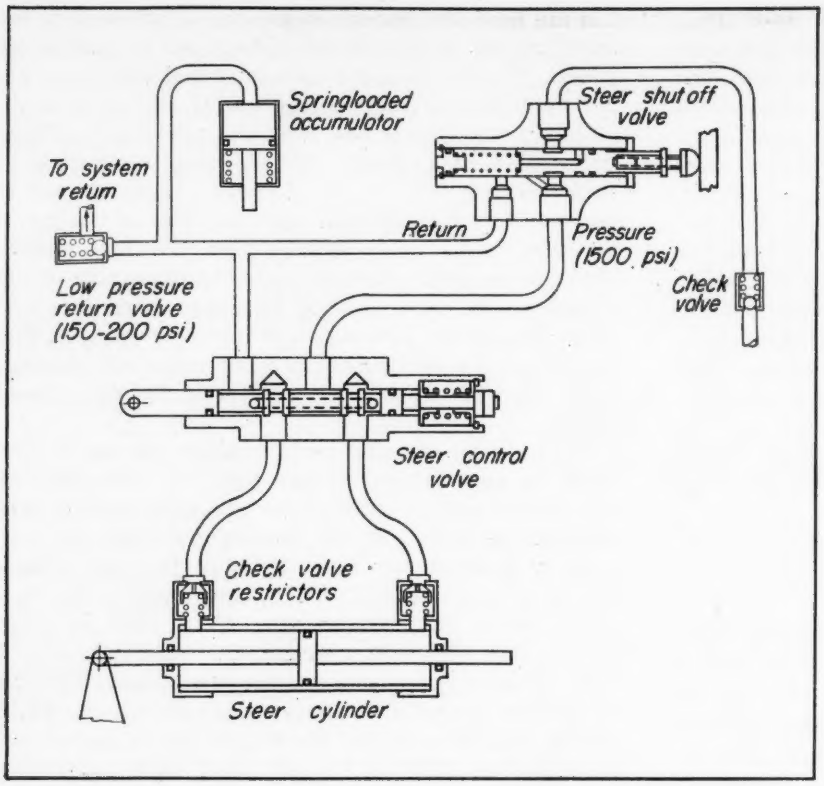



Fig. 6—Section view of hydraulic system for nose-wheel steering

return line and back to the reservoir. When the pilot releases the steer wheel, the centering spring acts and returns the valve to neutral. Steering is accomplished in the other direction in substantially the same manner.

From the foregoing, it can be seen that on the ground the spring-loaded accumulator would be kept full at all times by steering action and also by internal leakage from the pressure port in the steer valve. It is also necessary that this accumulator be kept full in flight to insure shimmy damp protection at the time of landing. As soon as the landing gear is raised after the take-off, the steer shutoff valve plunger moves to the right by spring action and blocks the system pressure. At the same time, the steer-valve pressure port is vented to the steer-valve return port through the steer shutoff valve. Thus, any leakage from the shutoff valve pressure port will act to keep the accumulator full in flight rather than build up pressure to the steer valve.

It can be seen that the steer shutoff valve provides protection against inadvertent steering at all times when the landing gear is retracted. Protection is also necessary, however, when the landing gear is extended with the airplane in flight to prevent landing with the nose wheel off center. This is accomplished mechanically in the oleo strut by means of a centering cam. This cam acts to center the wheel as the oleo extends and in the last increment of extension, actually locks the wheel in center. Thus, even if steering pressure is applied, the wheel cannot turn until the oleo is partially compressed by landing loads.

A small electric-motor-driven pump provides for emergency operation of the hydraulic system in case of failure of the main engine-driven pumps. The pressure line from this pump connects into the main pump presure line just before it enters the unloading valve, such that it is capable of operating both the 1500-psi and 3000-psi pressure circuits in the normal manner. The main and emergency pressure lines are checked off from one another at their point of connection. A separate source of fluid supply for the electric pump is maintained at all times by connection of its suction line to the bottom of the reservoir, whereas the main pump suction line connects to a standpipe in the reservoir. Thus, if a failure occurs in the main system pressure or suction lines and the normal system fluid is lost overboard, retention of sufficient fluid for operation of the system by the electric pump is assured.

An emergency by-pass valve is mounted on the floor convenient to the pilot. This valve is normally closed and would be opened only in case the open-center flow becomes blocked due to malfunction of the unloading valve, landing gear valve, or flap valve. Such a stoppage would cause continuous 3000-psi flow over the main relief valves and would result in overheating of the hydraulic fluid unless relieved by the emergency by-pass valve.

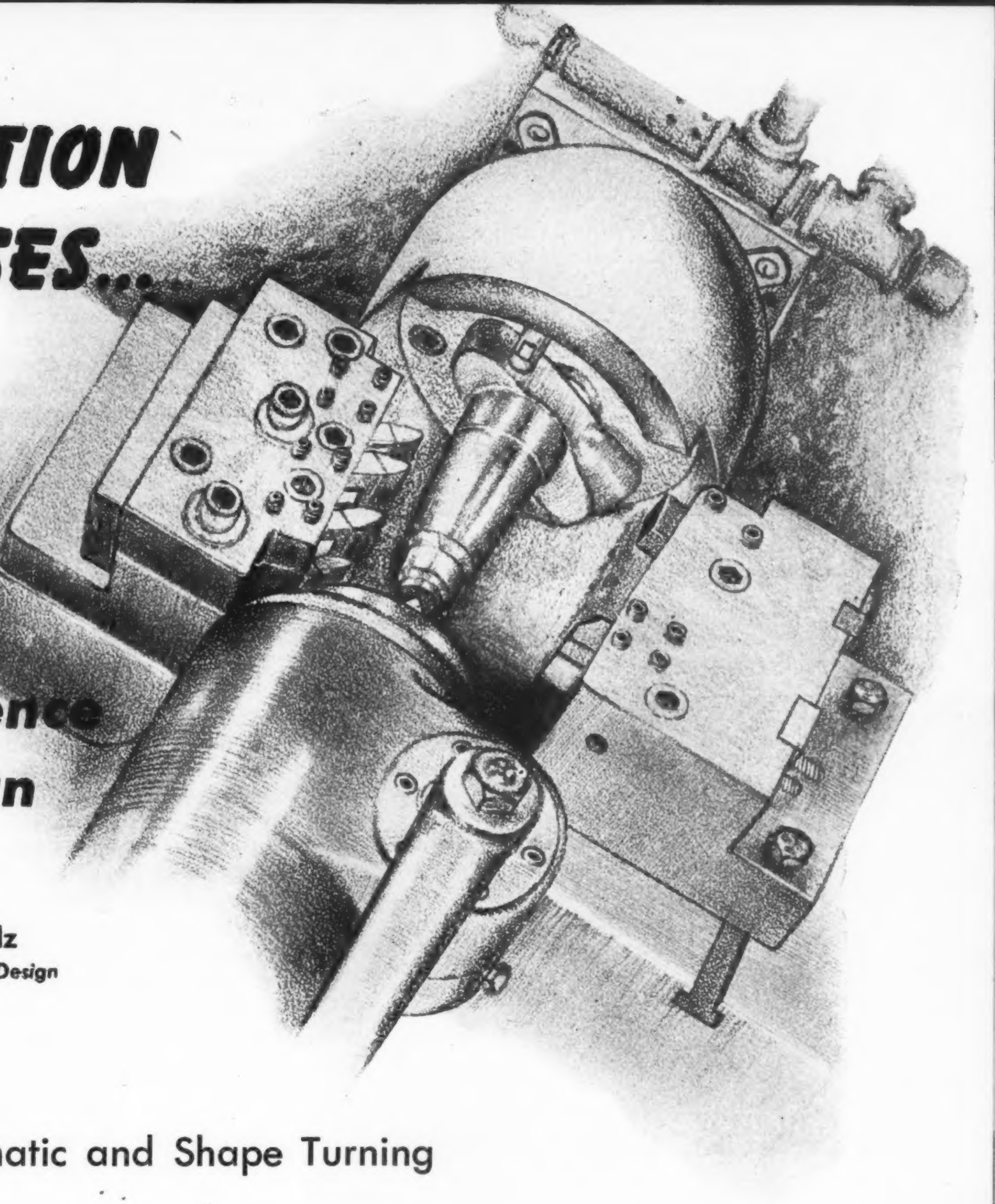
## Plastics-Supply Picture Brightening

IT now appears that 1948 will be the first year since prewar in which the supply of all major plastics materials will be somewhere nearly adequate to meet demand. The plastics industry today—expanded six-fold since 1939—is geared to produce and convert into finished goods more than a billion pounds annually of chemically-made basic materials.

Supplies of most thermosetting plastics materials during 1947 were not equal to requirements. This was particularly true of most molding powders, adhesives and many specialty materials. Most critical supply problem facing the thermosetting material manufacturers is that of formaldehyde, and current supplies of phenol and urea are not much better. Government reports indicate, however, that production of the needed raw materials has reached high levels and producers state the supply-demand ratio is rapidly approaching balance. In general, the thermoplastics are expected to be available in sufficient quantities to meet requirements. Certain types may be in short supply during the first quarter of the year but this should be overcome in the second quarter.

# PRODUCTION PROCESSES...

## Their Influence On Design



By Roger W. Bolz
Associate Editor, Machine Design

## Part XXXI—Automatic and Shape Turning

PLAIN turning operations, according to survey results, constitute the primary or basic machining functions employed in manufacture. The useful and highly desirable attributes of the engine lathe—which generally have been supplanted by those of the turret lathe and automatic screw machine wherever possible because of production economies—today are available in a wide variety of automatic lathes. All the advantages of single-point tooling for maximum metal removal, finish accuracy, center turning, etc., are now at the designers fingertips, with production speeds on a par with the fastest processing equipment on the scene today.

Turning equipment of this nature can be divided into two broad classifications. Full automatic, *Fig.* 1, and semiautomatic, *Fig.* 2, lathes constitute the first general category while full automatic, *Fig.* 3, and semiautomatic, *Fig.* 4, tracers or shape turners, *Fig.* 5, constitute the second. Tracers are available in either hydraulic or air-controlled models which employ low-cost templates for reproduction. Shape turners, however, employ a combination of axial electric tracing and mechanical tool stroke actuation with turning radius and stroke control. Automatic lathes in the first classification are available in a variety of standard or special models with mechanical, electrical or hydraulic actuation.

AUTOMATIC-CYCLE LATHES: Employing primarily only simple, single-point turning operations, automatic-cycle lathes are ideally suited to the production of a wide variety of machine parts such as automotive and aircraft pistons, aircraft engine cylinders, transmission cluster gear blanks and shafts, camshafts, bearing races, spur and bevel gear blanks, etc. Unlike the turret lathe and automatic screw machine, these lathes are adapted to the machining of forgings, castings and bars held between centers or work supported on fixtures as well as ordinary chuck work. Plain bores on many parts, preferably for accuracy in chucking, are best finished in a preliminary opera-

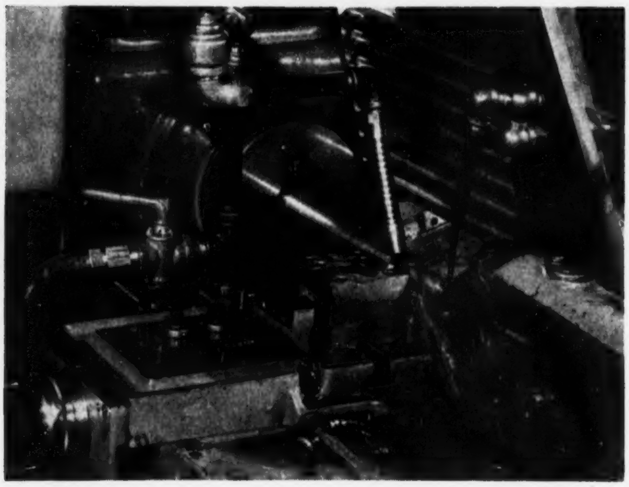
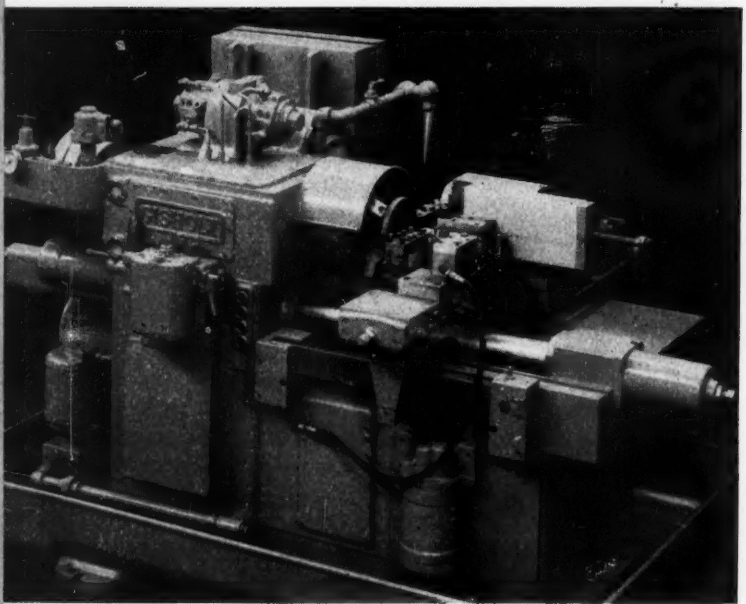
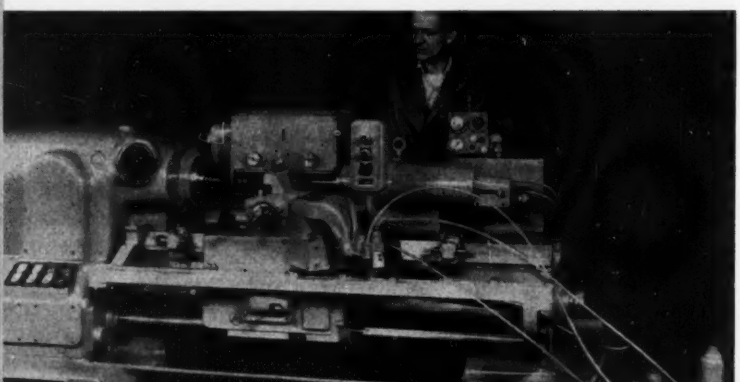


*Fig. 1 — Above — Arranged for fully automatic magazine loading, this automatic is one of the smallest available*

*Fig. 2—Below—Automatic lathe, arranged for hand loading, employs rocking tool slides and hydraulic actuation*





tion on a turret lathe or by drilling and broaching. Special bores or tapers, however, can often be more readily produced on automatic lathes owing to the advantage of pivoted tool posts, *Fig.* 6.

In general, parts which fall within the scope of automatic lathes would be:

1. Those which because of length or special requirements as to accuracy and finish can only be turned satisfactorily on centers, including work turned in preparation for grinding or other machining requiring the use of centers
2. Those which because of having been forged, cast or welded canont be cut from a bar or fed through a hollow spindle
3. Forgings, castings, etc., which, on account of irregular shape, cannot be readily held in chucking machines
4. Those which require machining all over or at both ends
5. Those parts which cannot be chucked and lend themselves to stacking for magazine feed
6. Those parts required in small quantity not within the economical range of automatic screw machines.

Where parts are long and require machining at both ends, a center drive attachment can be employed which acts both as steady rest and driver. Large, long parts which require machining all over are usually driven by spur drives, *Fig.* 7.

Automatic lathes range in size from small units designed to handle work ½-inch to 1⅜ inches in diameter by 2½ inches to 18 inches in length, to units capable of handling diameters up to 24 inches and lengths up to 8 feet or more. Powered by motors from 5 hp on the small units up to 75 hp on the larger machines, full advantage of carbide tooling can be obtained wherever practicable and maximum possible production speed assured. Automatic tool relief at the end of each tool stroke avoids scraping of the tools, increasing tool life and providing better finish.

Although lengths which can be turned are necessarily limited by the carriage travel in many machines to 12 inches or less, longer lengths of turn can be readily handled by multiple tooling. Truck steering knuckles, for instance, are produced in 0.9-minute or 66 per hour, *Fig.* 8. Small 9-inch compressor crankshafts, *Fig.* 9, are produced from a rough-turned blank in 0.56-minute or approximately 107 per hour. As a rule, floor-to-floor time averages around 30 seconds, ranging from a minimum of about 7 seconds (three seconds machining time) to around a maximum of 5 minutes.

Changeover of automatic lathes is relatively simple, few models requiring any new cams or equipment beyond the tooling for each new job. Setup time is short and flexibility is sufficient to encompass a broad field of design. In many cases design changes and improvements on parts being run involve little or no cost over setup.

AUTOMATIC TRACERS: Similar to the automatic-cycle lathes, automatic tracer lathes employ single-point tooling but, ordinarily, only one tool. Primary

*Fig. 3—Left—Fully automatic air tracer lathe loads, then single-point turns two diameters, a taper, an undercut, and a chamfer, and reloads*

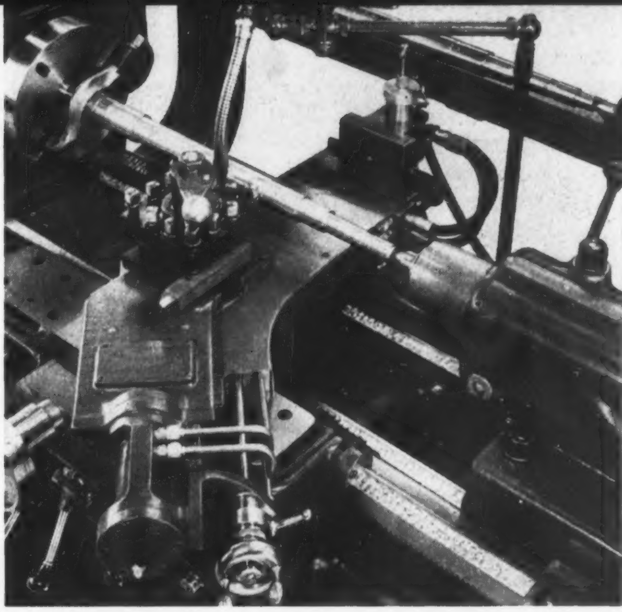


*Fig. 4—Right—Semiautomatic hydraulic tracer reproduces a variety of designs using an actual part as a pattern*

limitation to design is that all details must be possible to reach with a sharp-point tool. In general, parts which are considered suitable for obtaining the advantages of the automatic tracer are somewhat similar to those mentioned previously but usually contain a complicated form, taper, or other details which cannot be reproduced except with complicated banks of tools or special attachments. Characteristic parts are valve plugs, valve bodies, nozzles, orifices, stepped shafts, contoured rolls, etc.

Employing low-cost templates for reproduction, automatic tracers are ideally adapted to production quantities ranging from only a few pieces to large-quantity output. Compressor crankshaft, *Fig.* 10, has five stepped diameters, two-tapers, radii, and shoulders turned in 24 seconds from the rough casting. Automatic tracer lathes range in sizes paralleling those of the automatic cycle lathes and offer a broad coverage of machine parts.

AUTOMATIC SHAPE TURNERS: Rounding out the lines of automatic turning equipment are the automatic shape turners. These machines offer possibilities for design which cannot be obtained by any other means. Though originally developed for the manufacture of hubs, dies and molds of compound shapes, these machines are fast finding use in the efficient, economical production of parts for some of the newer, more complex machine parts, *Fig.* 11.

Much the same as with automatic tracers, only low-cost, thin sheet-metal templates are necessary. An almost infinite number of composite or compound shapes and contours are possible, the only limitation being that all details permit the entry of a single-point tool into the cut. Cross section can be round, square, triangular, oval, or any multisided figure having a maximum radius variation of not more than 2½ inches. Sections can be produced with as many as 520 flat, convex or concave flutes about their periphery. Relatively sharp corners can be maintained where required and sections such as fluted ovals can be blended smoothly into a circular section, carrying the flutes continuously or gradually eliminating them as desired.

Capable of handling parts up to about 20 inches in diameter, automatic shape turners like ordinary lathes are adaptable to handling either center or chucking work. Bored contours, *Fig.* 11, can be as readily produced as external ones and to a high degree of accuracy. Likewise, a multitude of facing designs can also be produced ranging from squares to extremely complicated contours, *Fig.* 12. Accurate and closely fitted mating components of complicated design can be readily produced in production, *Fig.* 5. Speed is somewhat lower than with the other lathes.

DESIGN FACTORS: Parts which are "naturals" for automatic lathe production, because of oné or more of the various factors outlined previously, can be designed to assure minimum production cost by observing a few basic rules. Naturally, production time is controlled to a great extent by the amount of turn-

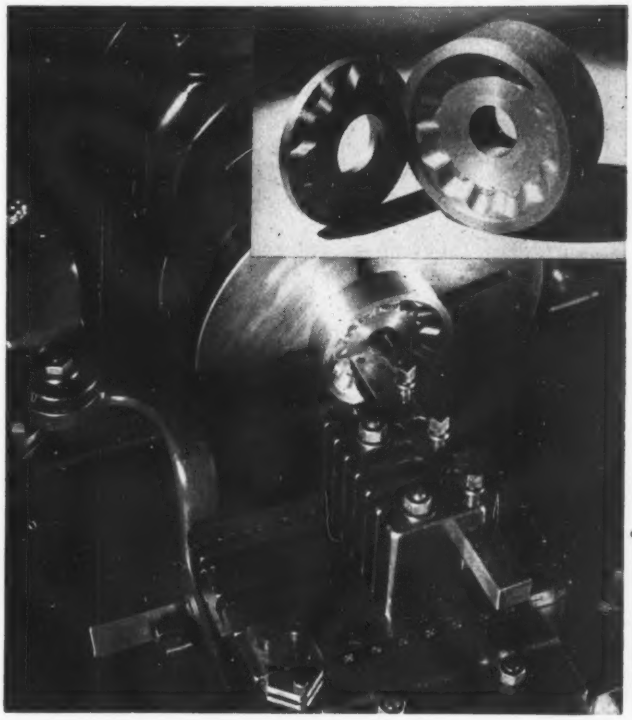


*Fig. 5 — Above — Automatic shape turner producing a tapered, dog-toothed part. Male and female pieces mate perfectly because of mechanical duplication of spacing and contours*

*Fig. 6—Below—Automatic lathe, turning inner groove of an outer ball bearing race, showing rocking tool slide*

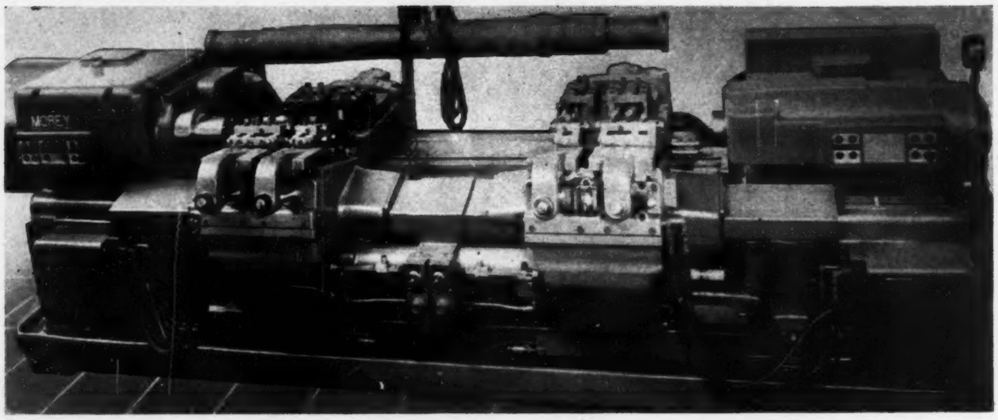
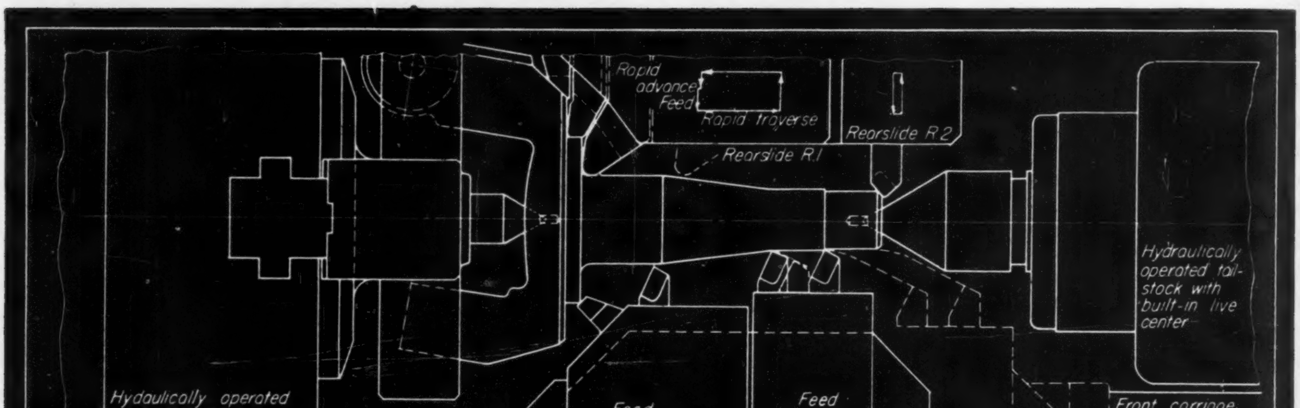


Fig. 7 — Left — One of the largest of the semiautomatic hydraulic lathes, this 150-hp axle lathe turns both ends simultaneously and drives the part by means of a spur center hydraulically loaded

Fig. 8—Below—Layout of tooling for steering knuckles. Production speed of 0.9-minute floor-to-floor with 0.6-minute machining time is attained




ing and consequently the turned portions of a part should be kept to a minimum wherever possible.

Second controlling factor is handling time. Owing to speed of turning operations, parts should always be considered for the possibility of hopper or chute feeds and automatic loading to attain maximum production. Such parts, of course, should be symmetrical to a degree, offer few problems as to orientation and be easy to chuck, grip with a collet, hold on an expanding mandrel, or center.

Economical design dictates easy chucking or holding for all parts as well. Complex holding devices often slow down operations drastically. Because one operator often handles several machines, chucking time is at a premium and this factor should always be kept in mind. Light or thin-wall parts often cannot be held by conventional means without distortion and unless some other means is readily adaptable, such as perhaps vacuum chucking, turning at optimum speeds and feeds may not be practicable. Again, long, slender parts may not allow maximum feeds and speeds and where production speed is a factor parts should be sufficiently stiff to withstand the cutting pressures and experience has proved that the extra metal is more than compensated for in production savings.

Wide forming tools such as are common with turret lathe and screw machine production are not employed on automatic lathes. Forms suitable for a single lathe tool are used but where long formed sections are required single-point generation is used. Generation beyond simple tapers and curves is not usually handled, and the automatic tracer or shape turner must be used.

Square shoulders and wide faces pose few problems owing to use of single-point tooling and automatic tool relief on return. Likewise, relatively sharp corners can be produced but to favor tool life and production, corner radii should be not less than 0.005-inch and preferably greater.

Necking or undercutting often presents a problem in automatic lathe production. As designed, the cluster gear, *Fig.* 13, requires that the undercutting tool face as well as neck. To prolong the usefulness

of the shortlived necking tool, the alternate design shown, which relieves the tool of this extra duty, is preferred. Function is not altered and especially since the neck is only for purpose of clearance in gear cutting the odd shape design is best eliminated.

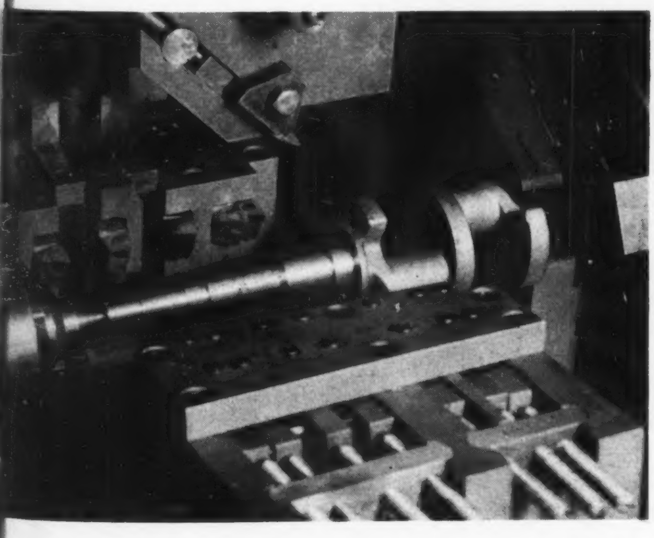
Quality of finish is important and at maximum production speed fine finish is not always possible. Small bores, for instance, usually cannot be produced without a mark when the tool is retracted. Tool relief overcomes this but cannot be used generally on bores under about 3 inches. Consequently, unless a tool mark is permissible, two operations must be employed, increasing cost and handling.

On external parts where maximum production is subordinate to quality of finish, *Fig.* 7, special attention to tooling can usually provide the desired results. The same situation is prevalent, however, in that fine finish usually requires two operations to assure elimination of tool marks. Occasionally, rough and finish turning operations can be done by proper tooling in one setup if the work permits the heavy cutting loads.

On sections which are to be finish ground after turning, only a rough turn is required and coarse feeds are desirable to reduce the abrasive action between the tools and the work. Ordinarily, about 0.015-inch is allowed for the grinding operation and a satisfactory rough turn can have a surface finish of about 400 microinches. Where finer finish is used, grinding allowances can be reduced on centered work to an amount just sufficient for removal of tool marks and, if hardening is involved, distortions. Careful design can thus make substantial savings in grinding time. On those portions of parts having no precise fit for mating, a fairly good finish should also be specified, possibly from 75 to 150 microinches.

MATERIALS: As with other turning operations, high machinability rating is an all important factor to be taken into consideration in specifying materials. A fairly complete table of machinability ratings is contained in Part III of this series and to insure maximum tool life and good finish materials with the high-

*Fig. 9—Compressor crankshaft finished from rough-turned blank in 0.56-minute with handling time of 0.1-minute*



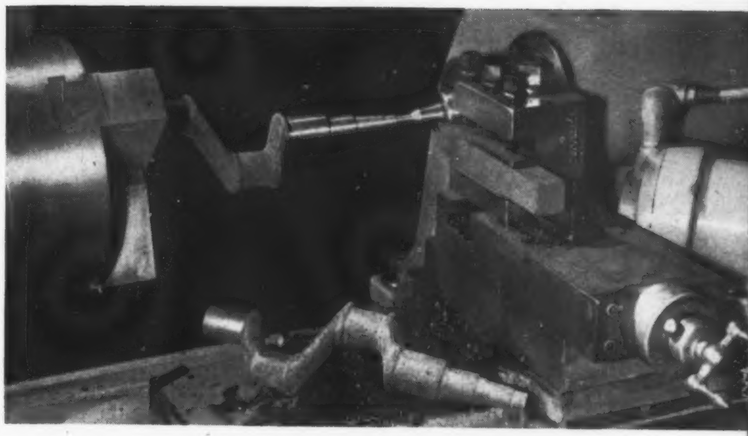
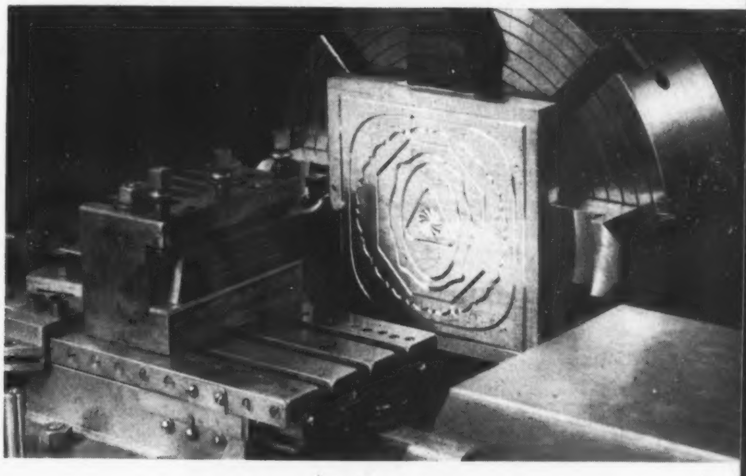


*Fig. 10—Above—Requiring no rough-turn operation, this automatic tracer completes a crankshaft in 24 seconds*

*Fig. 11—Right—Helical pump rotor and body section, both of which are produced on the automatic shape turner*

*Fig. 12 — Below — Sample plate showing a few of the facing designs which can be produced on the shape turner*



est possible machinability rating should be favored.

TOLERANCES: Surface finish depends greatly upon the material turned, tooling, and speeds and feeds employed. As mentioned previously on rough-turn operations, surface finish may run to 400 or more microinches with coarse feeds down to possibly 75 to 150 with finer feeds, *Fig.* 14. For light finish turning

operations on work such as bronze bushings, small cast-iron parts, or bearing races, *Fig.* 15, which require a fine finish with a polished appearance, surface quality around 30 microinches can be obtained. The railway axles in *Fig.* 7, for instance, must meet rigid finish requirements. The journal bearings are burnished after turning and no tool marks or chatters can be tolerated. Too, finish on the wheel seat must be suitable for press fitting and no burnishing by the tool can be tolerated.

Extremely close dimensional tolerances can be attained readily when desirable. Truck, *Fig.* 8, or automotive steering knuckles, for example, are held to plus or minus 0.002-inch in continuous production using but one cut. Automotive and aircraft pistons are produced with groove widths held within plus or minus 0.0002-inch.

In general, on high production runs where maximum output is desirable, a tolerance of plus or minus 0.005-inch is desirable on both diameter and length of turn. Where no fit or succeeding operation requires otherwise, tolerances of plus or minus 1/64-inch are conducive to low cost and high output speed. Where design necessitates, tolerances can be held to plus or minus 0.001-inch and on occasion plus or minus 0.0002-inch to 0.0005-inch.

Concentricities normally can be held to close limits especially when centers or precision bores are concerned. Ordinarily concentricity will be within a total dial reading of 0.002 to 0.003-inch, maximum.

Accuracy of work produced on automatic tracers parallels closely that of the automatic lathes and in many cases can be relied upon to produce complex contours with even smaller part-to-part variation. Compressor crankshafts, *Fig.* 10, for purpose of illustration, on lots of 2000 were within a maximum error of 0.0005-inch on all diameters.

Like accuracies are characteristic of the shape turners. The pump rotors, *Fig.* 11, are turned within plus or minus 0.001-inch on diameter and lead of the helix is such as to provide a medium fit with the bore of the pump body produced on the same type machine.

Collaboration of the following organizations in the preparation of this article is acknowledged with much appreciation:

American Tool Works (*Fig.* 4) ........ Cincinnati, Ohio
Gisholt Machine Co. (*Figs.* 2, 6, 14 and 15) Madison, Wis.
Jones & Lamson Machine Co. (*Fig.* 13) .. Springfield, Vt.
R. K. LeBlond Machine Tool Co. ...... Cincinnati, Ohio
Lodge & Shipley Co. ................ Cincinnati, Ohio
Monarch Machine Tool Co. (*Figs.* 3, 5, 10, 11 and 12) .......................... Sidney, Ohio
Morey Machine Co. (*Figs.* 7 and 8) .. New York, N. Y.
Pratt & Whitney Div. Niles-Bement-Pond Co. (*Fig.* 1) ...................... West Hartford, Conn.
Seneca Falls Machine Co. (*Fig.* 9) .. Seneca Falls, N. Y.
Sunstrand Machine Tool Co. .............. Rockford, Ill.

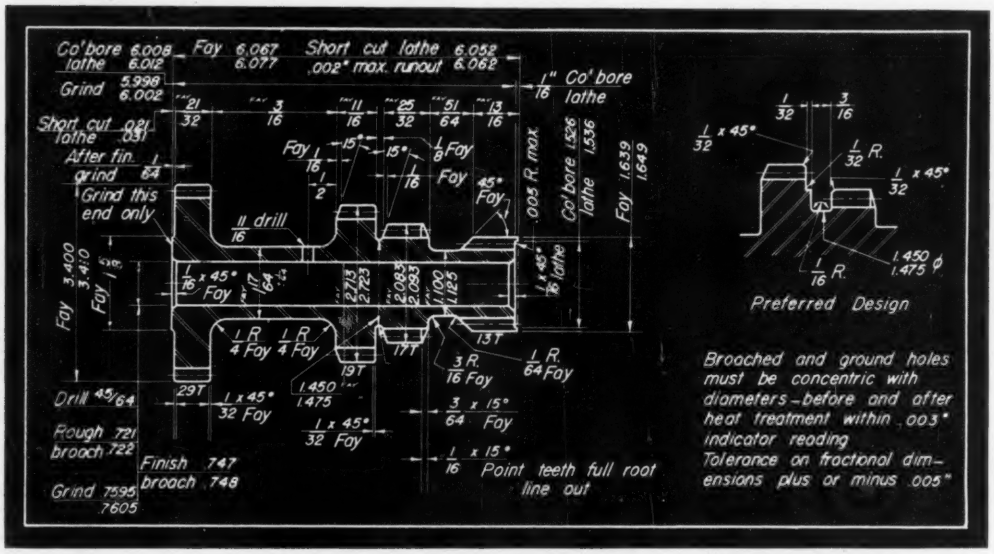



Fig. 13—Transmission cluster gear as designed for automatic lathe production and a suggested improvement for longer tool life

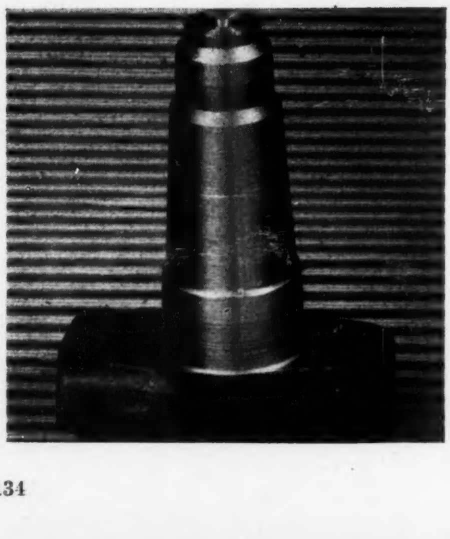
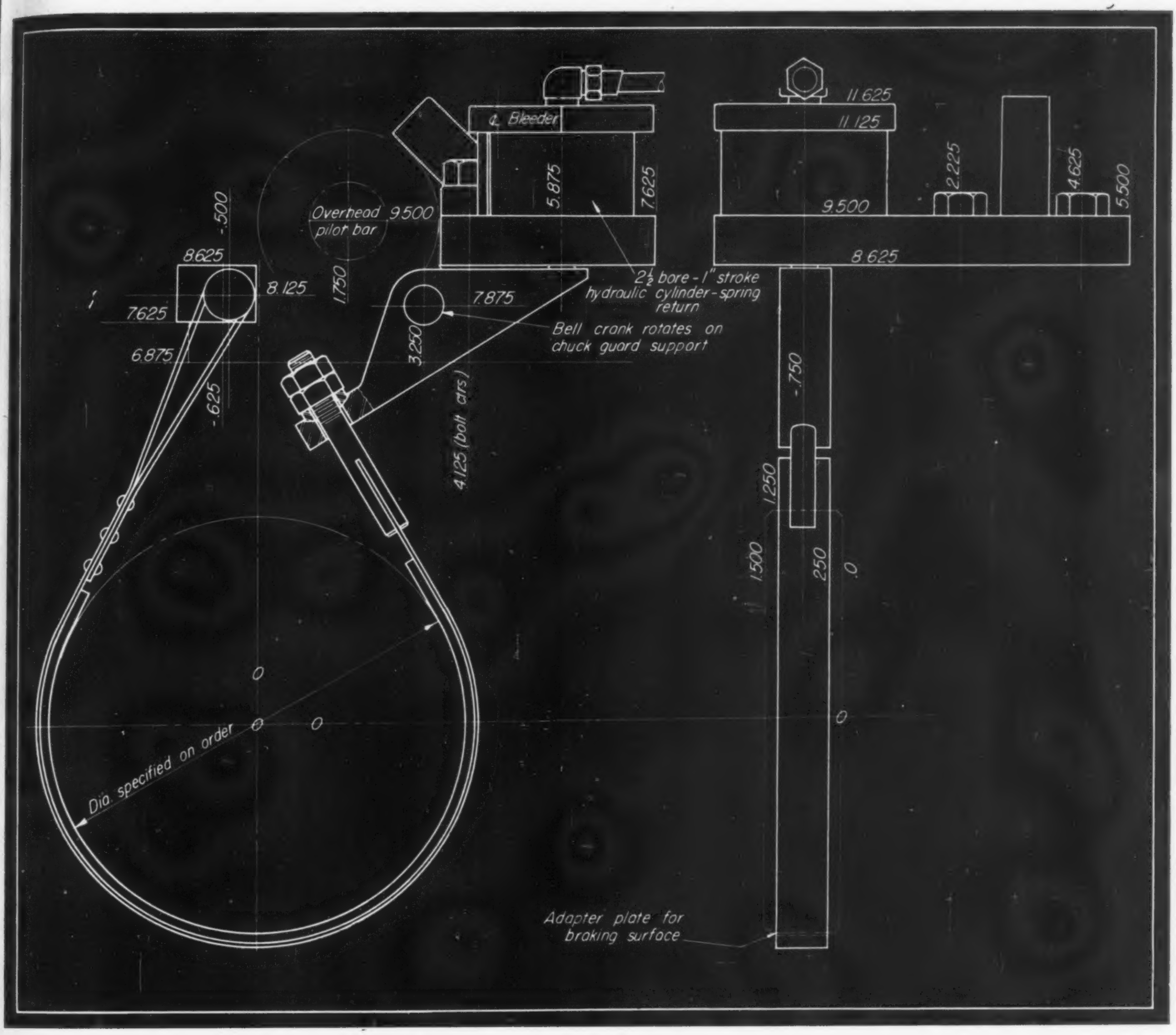


Fig. 14—Left—Diesel engine injector body showing quality of surface finish obtained in average automatic turning with a single operation and multiple tools

Fig. 15—Right—Inner and outer ball bearing races showing the high quality surface finish attained in finish turning operations




# Simplified Dimensioning System

**... based on Cartesian co-ordinates has many advantages over conventional methods used in machine layout**


By Norman W. Taylor
*Design Engineer*
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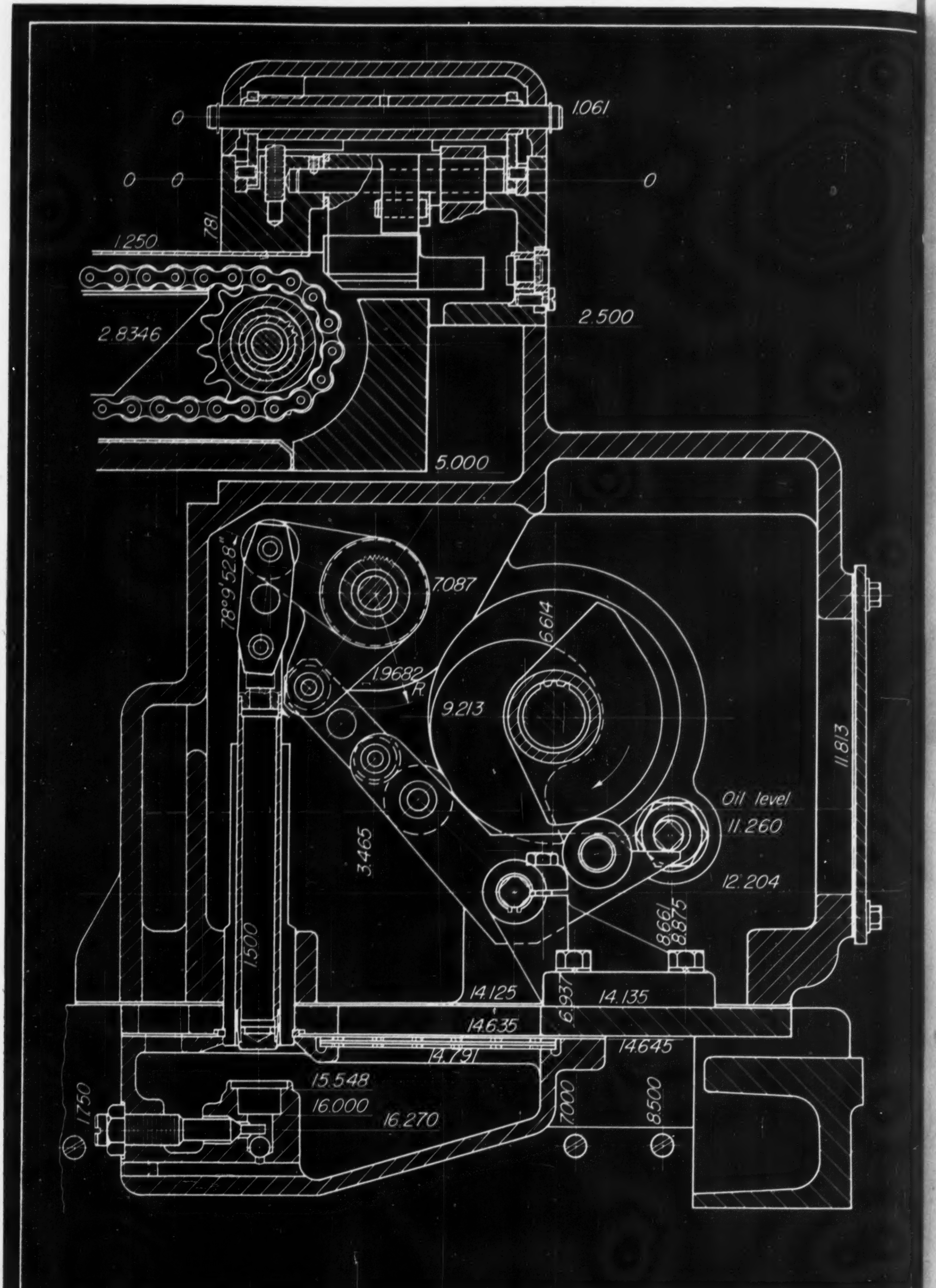
*Cleveland*


IN A manner of speaking, the engineer's language or method of communicating to others a new concept of any kind of mechanical design is, of course, the engineering drawing. Naturally, the clarity and accuracy with which such concepts are expressed is of utmost importance in proper interpretation. While this is particularly true of detail drawings, present accepted methods of detailing appear ade-

*Fig. 1—Above—Layout for a special lathe spindle brake showing basic principles of the Cartesian dimensioning*



1.061
0
0
0
0
.781
1.250
2.8346
2.500
5.000
78°9'52.8"
7.087
6.614
1.9682
R
9.213
11.813
Oil level
11.260
12.204
3.465
8.661
8.875
1.500
14.125
6.937
14.135
14.635
14.791
14.645
15.548
16.000
16.270
1.750
7.000
8.500

quate and especially since only one part is covered on a detail drawing as a rule, severe problems in attaining the necessary clarity and information are seldom encountered. The same cannot be said of layout drawings which often encompass an entire machine of considerable size.

That improved clarity and facile, accurate interpretation of machine design layouts would be well worth while is beyond dispute. The interesting fact is, however, that such a possibility is already a reality in the well established and proved but little known technique commonly referred to as the Cartesian dimensioning system. Long used by such outstanding European companies as Sulzer Brothers in Switzerland, and adopted recently by a few aircraft builders in the United States, experience with this system has proved that much time can be saved and many errors avoided in the development of mechanical designs.

A simple application of the Cartesian co-ordinate method developed originally by Rene Descartes in the seventeenth century, the system consists basically of noting every centerline and part boundary line of a mechanical design with its distance from a basic centerline. No additional dimension lines or dimension line projections are necessary and the resultant advantages are self-evident.

To apply the principle of Cartesian or rectangular co-ordinates, a point in the mechanism which is of importance to the whole machine is selected, such as the center of rotation of a drive shaft at the plane of a flange. Through this point two perpendicular lines in the plane of rotation (face of the flange) are chosen, one vertical and the other horizontal. Then, a third line perpendicular to those in the plane already chosen is selected. It will, of course, be the centerline of the shaft and it is at once apparent that in any right angle projection drawing only two of these can appear as lines, and are henceforth known as base lines. The third line, naturally, will be a point.

## Clearances Readily Indicated

As previously mentioned, any line which may be needed as a reference in later work, whether it be a center line or part boundary line, will be noted with its distance from either base line. Where two moving parts are close together the noted co-ordinates indicate readily the clearance allowed.

The value of the system is at once apparent in any follow-up work whether it be the co-ordination of several mechanisms by the supervising engineer or in the development of additions or alterations. It is important to note that the use of this system in construction layout requires no extra lines as in our usual dimensioning method. Recall that the usual dimension requires two extension lines with a double-ended arrow line.

*Fig. 2—Left—Portion of a complicated machine layout which illustrates the clarity and simplicity with which dimensions and precise locations can be indicated*

To illustrate a simple application of the principle, *Fig.* 1 shows a special spindle brake on a lathe. The designer was first required to establish the zero co-ordinates or base lines. These were provided in the *construction* layout of the lathe. The centerline of the spindle, and the horizontal and vertical lines in the plane of the spindle flange serve as the zero co-ordinates. Taking then the notations of finished surfaces, tapped hole centerlines, and other part boundary notations it was a simple matter to fill in the necessary mechanism to design the brake and mounting. All this *without any reference to detail drawings.* If it had been necessary to use detail drawings the designer would have required numerous detail prints and would have arrived at all the machine boundary and center lines only by multiple subtractions and additions.

Where the work is on opposite sides of the base lines, conventional practice makes use of positive and negative signs. If the position is not obvious, work to the left or below base lines takes a negative sign; that to the right or above is positive and is assumed positive without the sign.

## Has Place In Detail

Thus far only the fundamental possibilities in the application of the method have been touched upon. While this method is not often used in making up detail drawings, it will be recognized as the very system used on the jig borer and its use on certain parts may be advantageous. Consequently, though the system need not be used in dimensioning the details, it does have a place in the detail. If the machine construction shows the basic co-ordinates, then by including identifying co-ordinates of any detail, that detail can be located readily on the layout and assembled on the machine in its proper position without error and without any chance of incorrect orientation.

From the mathematical point of view there is a real advantage. The reader will recall that the rectangular co-ordinate system is the fundamental tool of analytic geometry. If then, any of the theorems from it are applied, they may be used to advantage in determining important timing and clearance features. This is not possible by conventional means without many added layouts on work with complicated machines.

## Introduction Causes No Confusion

The introduction of a new system is frequently discouraged because of the confusion it causes by being joined or used concurrently. The Cartesian method neither confuses the present method of dimensioning nor, as was said before, is it intended to supplant the standard method of dimensioning details, *Fig.* 2. It merely offers a greatly simplified means for recording the multitude of dimensions and locations established as a machine design layout develops, eliminates any possible confusion from paper stretch, and makes possible a better, more accurate master drawing or layout.

# Comparing Rubbers

***. . . with respect to their properties, mechanical applications and economies in production***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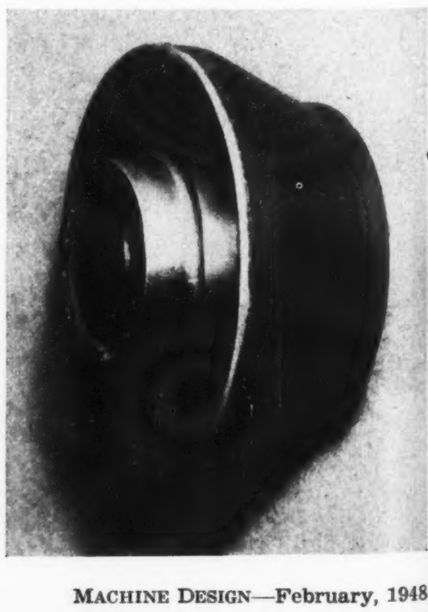
*Willoughby, Ohio*


*Fig. 1—Above—Pillow block is designed for use at elevated temperatures and under exposure to oil. Buna N is, consequently, a logical choice*

*Fig. 2—Below—Natural rubber is particularly suitable for the shear-type engine mount illustrated*



TODAY, designing rubber parts challenges the knowledge and ingenuity of the engineer as it could not have done fifteen years ago. Then, the engineer simply kept in mind the properties of one material—natural rubber. After he had specified the hardness and quality of the rubber (quality, by means of tensile strength and elongation at break) he could devote his efforts to designing the part. Now, however, before he can design a unit such as the pillow block shown in *Fig.* 1, he is confronted by many new materials having rubberlike properties.

Although none of the synthetic rubbers is equal to natural rubber in resilience, each has some physical or chemical property, TABLE I, which is superior to the analogous characteristic of natural rubber. Thus, synthetic rubbers have opened new fields in the production of mechanical equipment. Such items as rubber oil seals, cushion cups, gasoline hose, gasoline-pump diaphragms, inner tubes and linings impermeable to air, gases and chemical solutions could not have been made successfully without these recently developed materials.

NATURAL RUBBER: When properly compounded, natural rubber possesses good resistance to tear, abrasion, and "set" under compression or extension. Its resistance to aging, however, is inherently poor. When organic chemicals which act as antioxidants are added to natural rubber, the resistance to aging is improved to such an extent that many years of service can be expected.

The use of natural rubber in many products has been limited because of its poor resistance to deterioration in oils, solvents, heat,

and severe conditions of exposure to oxidation and ultraviolet light. Although the basic properties of rubber were most desirable in many applications, the use of natural rubber was precluded because of these limitations.

The fact that natural rubber has such high resiliency over a wide temperature range compared to synthetic rubbers is responsible for its use in making parts for equipment used under dynamic conditions but not required to withstand oil, solvents, or excessive heat. For this reason natural rubber is generally used for automotive and aircraft engine mounts, *Figs*. 2 and 3, and instrument mounts, vibration dampers, spring suspensions, bumper blocks, *Figs*. 4 and 5, and truck and bus tires.

Reclaimed Rubber: The tensile strength, elongation at break, and resistance to tear and abrasion of reclaimed rubber compounds are much lower than those of natural rubber. Resilience is approximately one-half that of natural rubber. The resistance of reclaimed rubber compounds to natural aging is not so good as that of natural rubber.

## Has Price Advantages

In many rubber products, where the physical qualities of natural rubber are not required, reclaimed rubber can be used to advantage economically. The material cost of reclaimed rubber is usually less than half that of natural rubber. During mastication, reclaimed rubber breaks down on the mill more readily than natural or synthetic rubbers and, consequently, requires considerably less power. It can be extruded on a tubing machine or calendered much faster and with a smoother control of gage. The uncured parts made of reclaimed rubber hold their shapes after extrusion and shrink less in cure than those made from natural or synthetic rubbers. Reclaimed rubber parts can be cured in two-thirds of the time required for curing those made of natural rubber and in less than half the time required for those made of synthetic rubbers. This is an advantage as it permits increased output from molds and press equipment. The costs of trimming, tumbling, and other finishing operations on parts made of reclaimed rubber are much less than those of natural or synthetic rubber.

Some idea of the importance of reclaimed rubber may be gained when it is realized that the output over the years has averaged about thirty per cent of the consumption of natural rubber. Approximately thirty pounds of reclaimed rubber are used in an automobile. Many rubber articles such as automotive mats, *Fig*. 6, running boards and step plates, *Fig*. 7, are made almost entirely of reclaimed rubber compounds.

Buna S (GRS): At room temperature GRS is approximately two-thirds as resilient as natural rubber. The loss of resiliency of GRS is greater at low temperatures (room temperature to 0 F) than that of natural rubber, but at high temperatures (room temperature to 200 F) the gain in resiliency of GRS is greater than that of natural rubber. Since GRS has less resiliency than natural rubber, GRS tires become much hotter, especially at high speeds. This eliminates GRS as a satisfactory material for the large tires used on trucks and busses.

The tensile strength of GRS compounds is much lower than that of natural rubber. Soft GRS compounds which measure less than 40 Durometer in hardness have a tensile strength of 200 psi; natural rubber compounds of the same Durometer hardness have a tensile strength of 2000 psi. The tear resistance of GRS compounds, especially at elevated temperatures, is considerably less than that of natural rubber. The resistance to set under compression or extension is very good; the resistance to loss on abrasion is also satisfactory.

## Should Not Be Used in Solvents

GRS should not be used in products where resistance to oil and solvents is required. Under most conditions the resistance to natural aging is comparable to that of natural rubber. However, special precautions should be taken in compounding GRS for use in parts such as automotive weatherstrips, *Fig*. 8, which will be used under tension and exposed to severe outside weathering conditions.

Summarizing, GRS does not equal natural rubber in many properties. In spite of its inade-

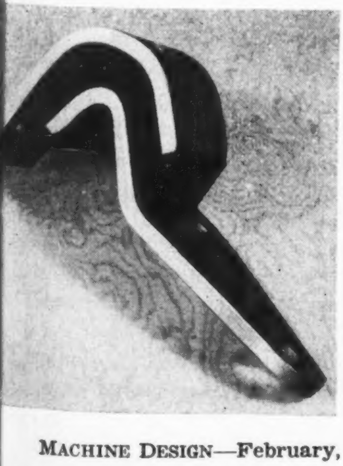


*Fig. 3—Left — Compression-type engine mount may be made of a number of rubberlike materials. Natural rubber was selected for the part shown*

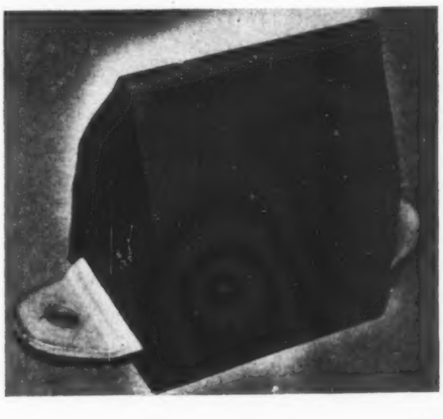


*Fig. 4—Right—Rear axle bumper for automobiles is made of natural rubber. Application of bumper is shown in Fig. 5*

quacies, however, GRS possessed enough quality to pull this country out of a bad situation in a national emergency. It will also prevent a severe strain on the national economy during the next few years by keeping the price of natural rubber from soaring to a price beyond reason.

BUNA N: At normal room temperature Buna N, like GRS, is approximately two-thirds as resilient as natural rubber. It stiffens considerably when cold and at 32 F loses much of its resiliency. At slightly below zero it becomes so stiff that it is brittle. However, if large amounts of organic plasticizers are used, the brittle point can be reduced to 40 F below zero. Buna N improves in resiliency at higher temperatures, 150 F to 200 F, as compared to the resiliency at room temperature. For this reason, together with the fact that it does not deteriorate as readily as natural rubber or most other synthetic rubbers at high temperatures, it was used during the war on aircraft for vibration insulators near the heat of the engine.

## Has Resistance to Set

The tensile strength of Buna N compounds is lower than that of natural rubber or Neoprene. Soft compounds of 40 Durometer of below, like those of GRS, test very low in tensile strength. However, the resistance of the compounds to set under compression or extension is very good. The fact that Buna N resists set under compression at temperatures of 200 F to 300 F, *Fig.* 1, much better than other synthetic rubbers or natural rubber was responsible for its use as aircraft and automotive gaskets and seals during the war.

The outstanding quality of Buna N is its resistance to swelling and to deterioration in oils, gasolines, and solvents. It has been used in large quantities for gasoline hose, oil and water-pump seals, and gasoline pump diaphragms.

Buna N does not resist oxygen or ozone as well as Neoprene. Specially compounded stocks made from the material resist aging at high temperatures (250 F—300 F) better than stocks made from any type of rubber except silicone.

BUNA N-VINYL BLENDS: Recently it has been found that Buna N and vinyl resins can be blended together. When small percentages of Buna N are mixed into vinyl resins, the Buna N acts as a plasticizer for the resin, thus making a thermoplastic which has excellent flexibility and which does not bleed plasticizer. This material has satisfactorily replaced natural and artificial leather in many applications.

When an equal amount of Buna N is mixed into vinyl resin, a new material is obtained which can be processed readily on rubber equipment and which can be cured into a thermosetting material. This new material has most of the resiliency and many of the physical properties of rubber in addition to luster and resistance to abrasion, chemicals, and natural aging.

NEOPRENE: At, or above, room temperature Neoprene GN is almost equal to natural rubber in resiliency, measuring much higher in this characteristic than any other commercial synthetic rubber. However, Neoprene does not remain resilient over as wide a range of temperatures as natural rubber. When measured at the freezing point of water, the resiliency of this synthetic is much lower than that of natural rubber. The physical properties of Neoprene compounds, including tensile strength and resistance to set, are very good at temperatures which are normally encountered.

Neoprene has not seriously threatened to replace natural rubber in general-purpose applications. The costs of material and elaborate manufacturing equipment, together with its tendency to stiffen in low temperatures, have been responsible for the limited use of the material. However, in industrial uses Neoprene possesses chemical properties far superior to those of natural rubber. Neoprene swells in petroleum oils and solvents, but does not disintegrate like natural rubber. It does not soften and deteriorate at high temperatures (150 F to 250 F) to the extent natural rubber does. Its outstandingly valuable properties are its resistance to aging at normal or elevated temperatures, and its greater resistance to the effect of oxygen, ozone, and ultraviolet light in comparison to other rubberlike materials, with the possible exception of butyl rubber. Like butyl rubber, Neoprene is relatively impermeable to air and many gases. Its weakest property in comparison to natural and most other synthetic rubbers is its inability to resist swelling in water.

The unusual properties of Neoprene have accounted for the opening of a new field of adaptations which

*Fig. 5—Application of the automobile rear axle bumper which is shown in Fig. 4*

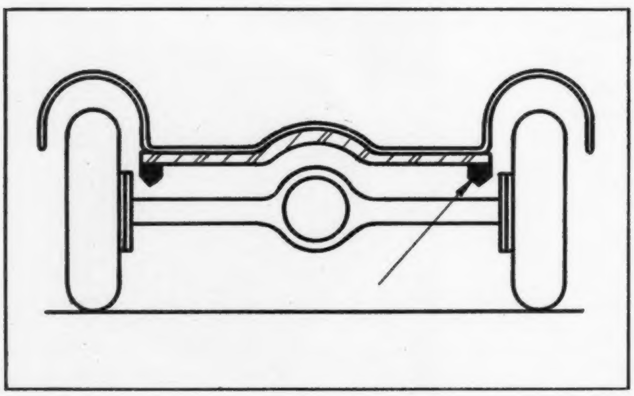
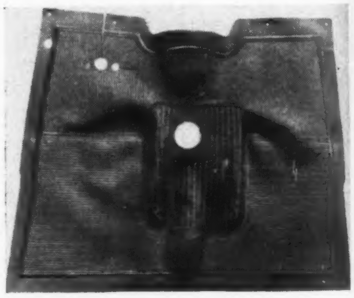


*Fig. 6 — Economies are realized in making automobile floor mats of reclaimed rubber*



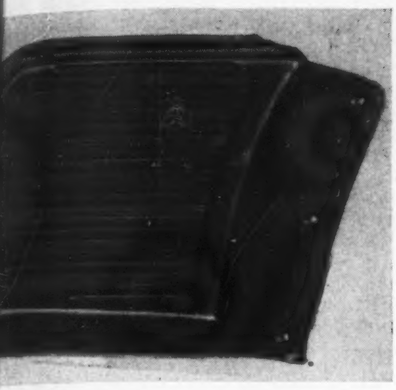


*Fig. 7—Step plate is reclaimed rubber bonded to a metal stamping*

did not exist when natural rubber was the only material available. Among the new products are oil-resisting engine mounts, motor cushion mounts, *Figs.* 9 and 10, oil seals and bearing-cushion cups.

Neoprene FR was developed to overcome stiffening at low temperatures, *Figs.* 11 and 12. The use of Neoprene FR is limited because other desirable properties, such as tensile strength and oil resistance, have been lowered in comparison to those of Neoprene GN. The resistance of GN compounds to low temperatures has been greatly increased within the last few years by the addition of special plasticizers.

**Butyl Rubber:** The physical properties of butyl compounds, including tensile strength, elongation, resistance to tear, resiliency and abrasion, are much lower than those of natural rubber compounds. Butyl compounds have very poor resistance to set under compression or extension and they stiffen considerably more than natural rubber compounds at cold temperatures.

Although butyl rubber is not resistant to oils and solvents, other chemical properties give it an advantage in special applications over other rubberlike materials. It is practically impermeable to air and many gases, making it useful for inner tubes, gas masks, etc. Its resistance to acids has developed its use in equipment for chemical plants.

Butyl rubber is a highly saturated material chemically. This means that its carbon bonds do not readily take up oxygen, sulphur, and other chemical elements. Consequently, butyl is highly resistant to oxidation and can be used where exceptional resistance to natural aging or ozone is encountered. However, because it is highly saturated chemically it cures very slowly; that is, it does not readily take up sulphur as well as oxygen. In order to cure butyl rubber it is necessary to tie up press equipment for a long time and to add a high percentage of expensive organic chemicals to accelerate cure. Unsaturated organic materials, such as crude rubber, other synthetic rubbers, reclaims, and softeners, cannot be added to butyl rubber because, by robbing it of its sulphur, they prevent it from curing.

**Thiokol:** The first of this rubber-like material, Thiokol A, appeared on the market in 1930. Although Thiokol A was the first oil and solvent resistant rubberlike material, it has practically disappeared be-

**Table I—Properties of Typical Rubbers**

| Material | Natural Rubber | Reclaim Rubber | GRS | Buna N | Neoprene | Butyl | Thiokol | Silicones |
|---|---|---|---|---|---|---|---|---|
| Derived From | Tree—Hevea Brasiliensis and other tropical plants | Tubes, tire carcasses, whole tires and mechanical scrap | Butadiene and Styrene | Butadiene and Acrilonitrile | Chloroprene | Isobutylene | Polysulphide | Silicon |
| Tradenames | — | — | GRS Hycar OS Buna S | Hycar OR Perbunan Butaprene Chemigum | GRM Neoprene GN Neoprene FR Neoprene E | GRI Flexon | GRP Thiokol A Thiokol BW Thiokol FA | Silicones Silastic |
| Available Forms | Latex, Solid | Solid | Latex, Solid | Latex, Solid | Latex, Solid | Solid | Solid, Powder, Dispersion | Solid, Paste |
| Processing Qualities | Excellent | Excellent | Good | Good | Good | Good | Good | Good |
| **Physical Properties:** Resiliency | Excellent | Fair | Fair | Fair | Very Good | Fair | Poor | Poor |
| Tensile Strength | Excellent | Fair | Fair-Good | Good | Very Good | Good | Poor | Poor |
| Water Impermeability | Excellent | Good | Good | Good | Fair | Very Good | Very Good | Good |
| Gas Impermeability | Good | Fair | Good | Good | Very Good | Excellent | Excellent | Good |
| Flexure Life | Excellent | Fair | Good | Good | Very Good | Very Good | Poor | Poor |
| Abrasion Resistance | Excellent | Fair | Excellent | Excellent | Excellent | Good | Poor | Poor |
| Tear Resistance | Very Good | Fair | Fair | Fair | Good | Fair | Poor | Poor |
| Heat Resistance | Fair | Fair | Fair | Good | Good | Good | Poor | Excellent |
| Cold Resistance | Very Good | Fair | Very Good | Fair-Good | Fair-Good | Good | Fair-Good | Excellent |
| Cold Flow Resistance | Very Good | Good | Very Good | Very Good | Fair | Fair | Poor | Poor |
| Weight (lb/cu in.) | 0.0335 | 0.036-0.057 | 0.0338 | 0.036 | 0.045 | 0.0328 | 0.048-0.057 | 0.058-0.076 |
| **Chemical Resistance:** Natural Aging | Fair | Fair | Good | Good | Excellent | Excellent | Excellent | Excellent |
| Ozone | Poor | Poor | Poor | Poor | Excellent | Excellent | Excellent | Excellent |
| Light | Fair | Fair | Fair | Fair | Excellent | Excellent | Excellent | Excellent |
| Petroleum Oils | Poor | Poor | Poor | Excellent | Good | Poor | Excellent | Good |
| Aromatic Gasoline | Poor | Poor | Poor | Good | Poor | Poor | Excellent | Poor |
| Petroleum Gasoline | Poor | Poor | Poor | Very Good | Fair | Poor | Excellent | Poor |

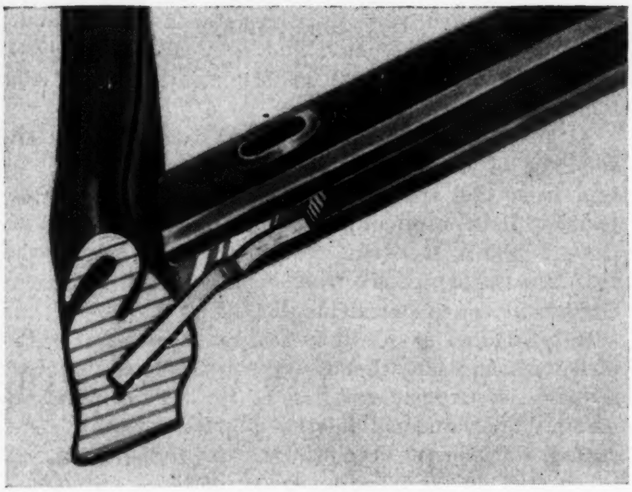


*Fig. 8 — Left — Automotive windshield weatherstrip is GRS synthetic rubber*

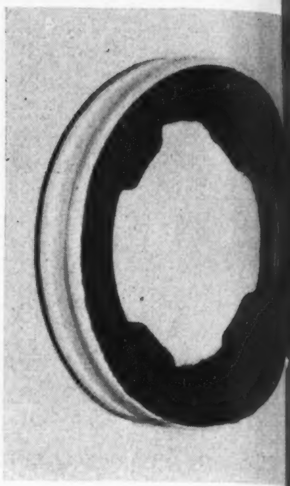


*Fig. 9—Right—Torque cushion mount for fractional-horsepower electric motor is made of Neoprene GN. Application is shown in Fig. 10*

cause of the very disagreeable gas given off during processing.

Thiokol FA, which has replaced Thiokol A, is an improved material. It does not "gas" in the factory and only a slight odor may be noticed in the finished product. Physical properties, including tensile strength, elongation at break, tear, and abrasion resistance, are poor compared to natural and most synthetic rubbers. The resistance to set under compression is poor because Thiokol FA is thermoplastic. However, there has recently been developed a Thiokol polymer which cross links during vulcanization. This new polymer, when vulcanized, has a high resistance to compression set.

## Used for Oil-Resistant Parts

The property which makes Thiokol rubbers practically indispensable in some industries is its resistance to swelling and deterioration in oils, gasolines, and solvents. Thiokol is also very resistant to oxidation, ozone and natural aging. It is impermeable to liquids and much less permeable than natural rubber to most gases.

Thiokol rubbers are used for the manufacture of such items as paint and lacquer hose, lacquering and other industrial rollers, solvent-resistant molded parts and coated fabrics.

Silicone Rubbers: Physical properties of the silicone rubbers, including tensile strength, elongation at break, abrasion and resistance and resistance to compression set, are poor compared to natural and most synthetic rubbers. For gasket applications, the compression set can be improved by severe baking in a high-temperature oven. This, however, can only be done at the sacrifice of other physical properties. The resiliency of silicone rubbers is very low, less than twenty per cent of that of natural rubber. However, the material maintains its resiliency over a wide range of temperature.

Commercially, the most important property of silicone rubber is resistance to deterioration in heat. Whereas continued exposure to temperatures of 300 F to 350 F quickly deteriorates or hardens natural and synthetic rubber compounds, continuous use at such high temperatures produces negligible change in flexibility and surface hardness of silicone rubber. Moreover, it does not become tacky at this temperature. In some applications the material is serviceable at a temperature of 500 F above zero; also it does not become brittle at 70 F below zero.

## Has Excellent Aging Resistance

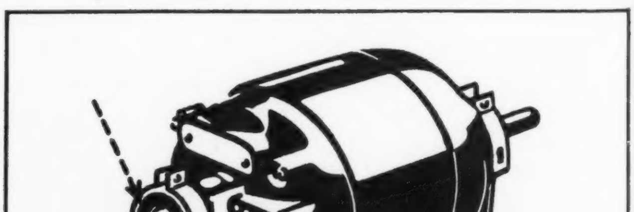
Silicone rubber has very good resistance to natural aging, oxidation and ozone. Although it is not classed as an oil-resisting material, it does not deteriorate in most oils. It does, however, deteriorate in gasoline.

The use of silicone rubber has been limited by the high cost of the material. At a pound cost of four dollars and a specific gravity of two, the volume cost of silicone rubbers is from twenty to fifty times that of natural and most synthetic rubbers.

Processing: Synthetic rubbers can be processed in the same equipment that has been used for natural rubber. However, the resultant costs of processing are higher in many cases. Synthetic rubbers do not plasticize on the mill as readily as natural rubber, they generate more heat, generally require a longer time to cure, do not flow and knit in the mold, and tear more easily when hot.

*New Methods of Mixing:* When a high percentage of carbon black is mixed into natural rubber, GRS, or other synthetic rubbers, a great amount of heat is

*Fig. 10—Application of Neoprene GN torque-cushion mount to fractional-horsepower electric motor*



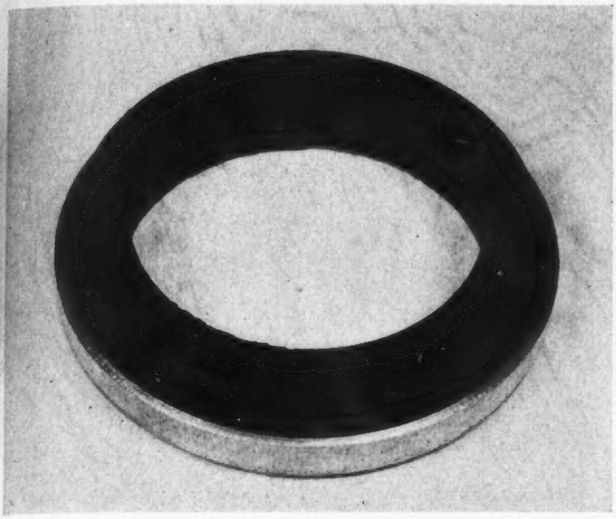


*Fig. 11—Dirt and mud seal for tractor front wheels. Seal body is Neoprene sponge to provide uniform pressure while seal face is Buna N to provide low friction and good wear characteristics. Seal is used as shown in Fig. 12*

produced. In the past, to make high-carbon GRS compounds workable, it was necessary to change to softer types of carbon blacks and to add larger percentages of plasticizers.

Recently it has been found that master batches can be made by incorporating the carbon black into synthetic rubber latex and coagulating the resultant dispersion. Not only has it been possible to incorporate higher percentages of carbon black into the synthetic, but the resultant compound has improved physical properties due to better dispersion of the carbon.

## High-Speed Manufacturing Method

*Injection Molding:* The Cousino injection molding machine designed to handle thermosetting plastic materials, particularly rubbers, is a screw type of injection machine, curing in a closed mold. Two rotating gears provide constant and positive pressure up to 20,000 psi on the material extruded by the screw. This pressure not only quickly fills the mold, but also preheats the rubber compound. When the mold is filled, the injection head is automatically lifted. This allows the compound at the nozzle to cool instantly and prevents scorching of the next charge into the mold.

The preheated rubber compound in the mold is cured in a very short time in platens heated to a much higher temperature than that used in compression molding. The advantage of the injection molding method over compression molding is that it is possible to obtain the same rate of production with fewer mold cavities. Also, it is possible to mold the hot plastic compound around inserts with great ease and with the probability of more consistent adhesion of rubber to metal since the mold is closed when the material is being injected. Too, the misplacement of inserts is materially reduced. The incoming rubber compound is uniformly heated and successive waves of disturbance caused by gradual closing of the mold are eliminated, thus preventing flow-lines and poorly

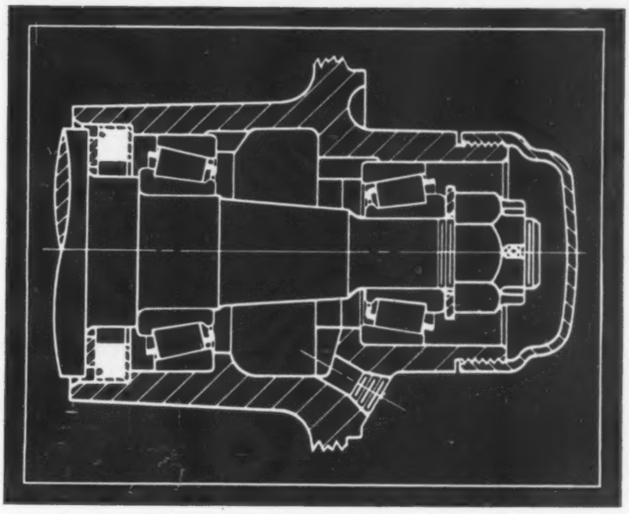


*Fig. 12—Installation of seal shown in Fig. 11 in tractor wheel hub. Rubber part is indicated in solid white*

knit production parts. Physical qualities of the rubber compound in injection-molded parts are usually better than those of comparable parts made by compression molding.

*Transfer Molding:* The transfer process can be used to advantage in the molding of rubber parts of large and intricate shapes or in the molding of parts requiring fragile inserts or sections. Transfer molding differs from compression molding in that the rubber, instead of being loaded directly into the mold cavity, is first subjected to heat and pressure in a transfer chamber. From the transfer chamber the rubber, in a plastic state, is forced through an orifice directly into the closed mold.

USE OF SPECIFICATION TABLES: Rubber Standards are now published in the SAE handbook under Section 4 (Fabricated Materials) and in the ASTM publication under tentative specifications D 735-43T. There are five tables. Three of these (S Tables) cover oil-resisting applications and two (R Tables) cover non-oil-resisting general-purpose applications for rubber compounds. In each table the grade numbers for the compound consist of three digits. The first digit indicates the Shore durometer hardness and the second and third indicate the quality as determined by tensile strength. The prefix letters indicate the classifications of the compounds: SA, maximum; SB, good; and SC, medium-oil-resisting compounds; RN, natural rubber or reclaim; RS, synthetic rubber non-oil-resisting compounds. The suffix letters are used, singly or in combination, after any grade number to indicate additional requirements beyond those specified in the basic requirements. In addition to the tables product specifications covering rubber fabricated materials, including rubber hose, hydraulic brake cups, windshield wipers, etc., are published in the SAE handbook.

In conclusion, the author wishes to stress the necessity for the designer of rubber parts to consult the published technical data on natural and synthetic rubber. If there are not sufficient published data at hand the designer should consult the source of supply to learn the most appropriate rubber to use.

# Disk Stresses

### *A practical method for calculating stresses due to rotation and to interference fits*

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THERE are various methods for determining the stresses due to rotation in disks. The sum and difference method has been described[1] and used widely. Its main shortcoming is that the method gives the disk stresses due to rotation only. To determine the bore stresses due to a shrink or press fit some other just as laborious method must be used to determine the relation between the radial and tangential bore stress for a stationary disk.

Methods involving a hyperbolic profile are not applicable to all disks. Where a hyperbolic profile method can be used it is usually confused by a hub and the fact that it is not practical nor economical to machine the profile to the shape used in the calculation. This method too is, in general, more laborious than the method discussed in this article.

The method to be described was partly suggested by a recent paper by K. E. Bisshopp[2], and the mode of attack is similar to that of the sum and difference curves. The "dead" load of the blades and rim is calculated and the radial stress is determined at some line above which there is no tangential load-carrying ability of the disk. Below this line the disk is divided into several convenient parallel-sided disks which approximate the disk's shape.

Referring to the Nomenclature on Page 148, for a parallel-sided disk, the well known general formulas can be written:

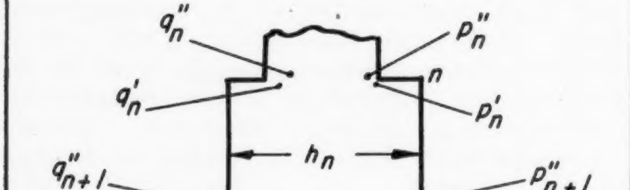
$$p = A - \frac{B}{r^2} - \frac{1}{8}(3+\mu)\rho\omega^2 r^2$$

$$q = A + \frac{B}{r^2} - \frac{1}{8}(1+3\mu)\rho\omega^2 r^2$$

Considering the $n$th section as shown on *Fig.* 1 and letting $p_n'$ and $q_n'$ represent the radial and tangential stress just at the start of the section of $h_n$ width,

[1] References are tabulated at end of article.

*Fig. 1—First step in analysis is to divide disk into several parallel-sided disks approximating actual shape*



$$p_n' = A - \frac{B}{r_n^2} - \frac{1}{8}(3+\mu)\rho\omega^2 r_n^2$$

and

$$q_n' = A + \frac{B}{r_n^2} - \frac{1}{8}(1+3\mu)\rho\omega^2 r_n^2$$

Adding the foregoing stresses,

$$p_n' + q_n' = 2A - \frac{1}{2}(1+\mu)\rho\omega^2 r_n^2 \qquad (1)$$

and by subtracting them,

$$p_n' - q_n' = -\frac{2B}{r_n^2} - \frac{1}{4}(1-\mu)\rho\omega^2 r_n^2 \qquad (2)$$

Just at the end of the section $h_n$ width,

$$p_{n+1}'' = A - \frac{B}{r_{n+1}^2} - \frac{1}{8}(3+\mu)\rho\omega^2 r_{n+1}^2 \qquad (3)$$

$$q_{n+1}'' = A + \frac{B}{r_{n+1}^2} - \frac{1}{8}(1+3\mu)\rho\omega^2 r_{n+1}^2 \qquad (4)$$

the sum and difference of Equations 3 and 4 are

$$p_{n+1}'' + q_{n+1}'' = 2A - \frac{1}{2}(1+\mu)\rho\omega^2 r_{n+1}^2 \qquad (5)$$

$$p_{n+1}'' - q_{n+1}'' = -\frac{2B}{r_{n+1}^2} - \frac{1}{4}(1-\mu)\rho\omega^2 r_{n+1}^2 \qquad (6)$$

*Fig. 2—Actual and approximate shape of alloy steel disk which serves as example illustrating the procedure*

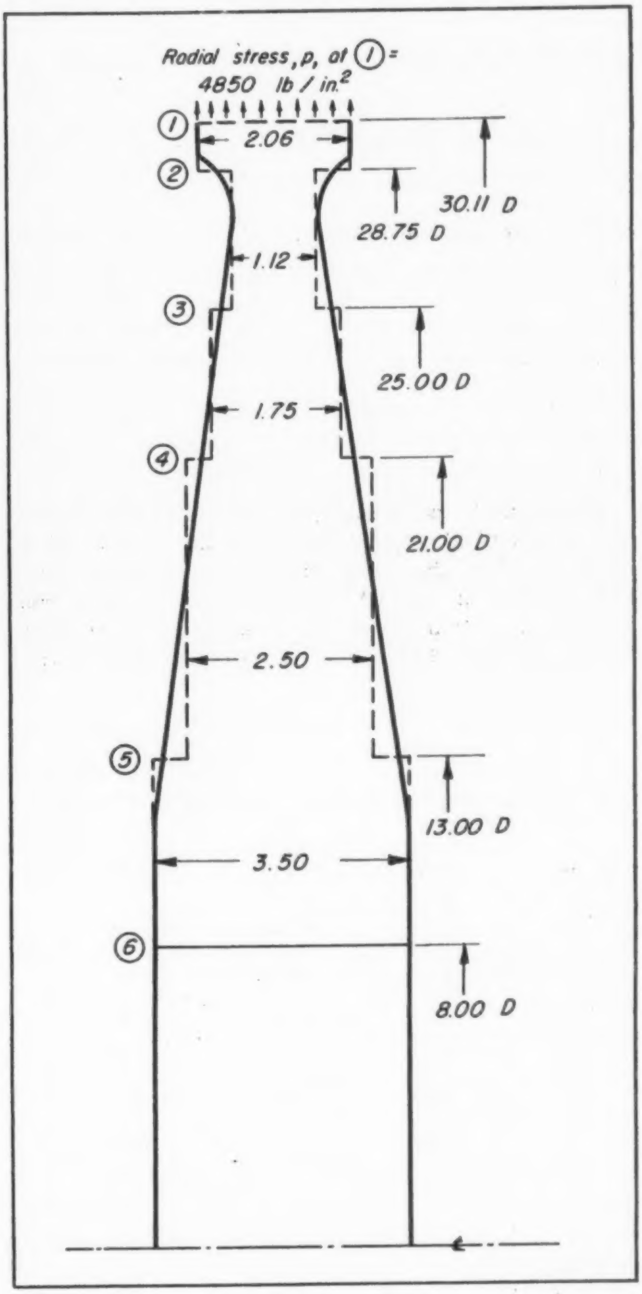


**TABLE I—Evaluation of Constants for Disk**

$N = 3987$ rpm $\quad \mu = .30 \quad \rho = 7.33\times10^{-4} \quad \omega = 418$ rad per sec $\quad \omega^2 = 17.5\times10^4 \quad [A] = \frac{1}{2}\rho(1+\mu)\omega^2 = 83.3 \quad [B] = \frac{1}{2}\rho(1-\mu)\omega^2 = 22.4$

| | [C] | [D] | [E] | [F] | [G] | [H] | [J] | [K] | [L] | [M] |
|---|---|---|---|---|---|---|---|---|---|---|
| $n$ | $r_n$ | $h_n$ | $\frac{h_{n+1}}{h_n}$ | $\mu\left(\frac{h_{n+1}}{h_n}-1\right)$ | $r_n^2$ | $r_n^2-r_{n+1}^2$ | $\frac{r_n^2}{r_{n+1}^2}$ | $[H]\times\{1+[J]\}$ | $[A]\times[H]$ | $[B]\times[K]$ |
| 1 | 15.06 | | | | 227 | 20.5 | 1.10 | 43.0 | 1710 | 963 |
| 2 | 14.375 | 2.06 | 0.545 | −0.1365 | 206.5 | 50.3 | 1.32 | 116.6 | 4190 | 2610 |
| 3 | 12.50 | 1.125 | 1.555 | 0.1665 | 156.2 | 45.9 | 1.417 | 111.3 | 3830 | 2490 |
| 4 | 10.50 | 1.75 | 1.428 | 0.1283 | 110.3 | 68.1 | 2.62 | 246.0 | 5670 | 5520 |
| 5 | 6.50 | 2.50 | 1.400 | 0.120 | 42.2 | 26.2 | 2.63 | 95.1 | 2180 | 2130 |
| 6 | 4.00 | 3.50 | | | 16.0 | | | | | |
| 7 | | | | | | | | | | |
| 8 | | | | | | | | | | |

Subtracting Equation 1 from 5 and rearranging,

$$p_{n+1}''+q_{n+1}''=p'+q_n'+\frac{1}{2}(1+\mu)\rho\omega^2(r_n^2-r_{n+1}^2) \qquad (7)$$

If Equation 2 is multiplied by $\frac{r_n^2}{r_{n+1}^2}$, subtracted from Equation 6, and rearranged,

$$p_{n+1}''-q_{n+1}''=(p_n'-q_n')\frac{r_n^2}{r_{n+1}^2}+\frac{1}{4}(1-\mu)\rho\omega^2(r_n^2-r_{n+1}^2)\times\left(1+\frac{r_n^2}{r_{n+1}^2}\right) \qquad (8)$$

Equations 7 and 8 will give the radial and tangential stress at the inner radius of the section of $h_n$ width provided the radial and tangential stress at the outer radius of this section are known. The transition from section to section is made in the conventional manner. The radial stress is assumed to vary inversely as the thickness at the transition, hence

$$p_{n+1}'=\frac{h_n}{h_{n+1}}\times p_{n+1}'' \qquad (9)$$

The change in tangential stress at the transition from one thickness to another is then equal to Poisson's ratio times the change in the radial stress; then

$$q_{n+1}'=q_{n+1}''-\mu\left(\frac{h_{n+1}}{h_n}-1\right)p_{n+1}' \qquad (10)$$

Equations 7, 8, 9, and 10 now give enough tools to calculate the stress distribution in a disk. The tables in which a sample calculation is listed can be readily filled out. In the solution for the stress distribution in a disk with a free bore two conditions are known: One is the radial stress at the outer diameter while the other is the radial stress at the bore. The latter, of course, is zero, while the stress at the outer radius is the stress due to the radial load above the diam-

**TABLE II—Trial Conditions, for Peripheral Stresses 4850 psi Radial and 10,000 psi Tangential, at Working Speed**

| | [N] | [P] | [Q] | [R] | [S] | [T] | [U] | [V] |
|---|---|---|---|---|---|---|---|---|
| $n$ | $p_n'$ <br> $\frac{[U]}{[E]}$ | $q_n'$ <br> [V]— <br> [F]×[N] | $p_n'+q_n'$ <br> [N]+[P] | $p_n'-q_n'$ <br> [N]—[P] | $p_n''+q_n''$ <br> [Q]+[L] | $p_n''-q_n''$ <br> [R]×[J] <br> +[M] | $p_n''$ <br> $\frac{[S]+[T]}{2}$ | $q_n''$ <br> $\frac{[S]-[T]}{2}$ |
| 1 | 4850 | 10000 | 14850 | —5150 | 16560 | —4700 | | |
| 2 | 10900 | 12120 | 23020 | —1220 | 27210 | 1000 | 5930 | 10630 |
| 3 | 9070 | 11600 | 20670 | —2530 | 24500 | —1100 | 14105 | 13105 |
| 4 | 8200 | 11750 | 19950 | —3550 | 25620 | —3780 | 11700 | 12800 |
| 5 | 7800 | 13760 | 21560 | —5960 | 23740 | —13520 | 10920 | 14700 |
| 6 | | | | | | | 5110 | 18630 |
| 7 | | | | | | | | |
| 8 | | | | | | | | |

**Table III—Conditions for Peripheral Tangential Stress of 10,000 psi at Zero Speed**

| | [N] | [P] | [Q] | [R] | [S'] | [T'] | [U] | [V] |
|---|---|---|---|---|---|---|---|---|
| $n$ | $p_n'$ | $q_n'$ | $p_n'+q_n'$ | $p_n'-q_n'$ | $p_n''+q_n''$ | $p_n''-q_n''$ | $p_n''$ | $q_n''$ |
| | $\frac{[U]}{[E]}$ | [V]—[F]×[N] | [N]+[P] | [N]−[P] | [Q] | [R]×[J] | $\frac{[S']+[T']}{2}$ | $\frac{[S']-[T']}{2}$ |
| 1 | 0 | 10000 | 10000 | —10000 | 10000 | —11000 | | |
| 2 | —918 | 10380 | 9460 | —11300 | 9460 | —14900 | —500 | 10500 |
| 3 | —1750 | 12470 | 10720 | —14220 | 10720 | —20150 | —2720 | 12180 |
| 4 | —3300 | 15860 | 12560 | —19160 | 12560 | —50200 | —4715 | 15435 |
| 5 | —13440 | 32990 | 19550 | —46430 | 19550 | —122000 | —18820 | 31380 |
| 6 | | | | | | | —51225 | 70775 |
| 7 | | | | | | | | |
| 8 | $k=-\frac{-51225}{70775}=0.724$ | | | | | | | |

eter where the calculation begins. For a solution these end conditions must be met.

In the use of the sum and difference curves the usual practice is to assume a tangential stress at the outer diameter and work inward to the bore. If the radial stress at the bore is not zero then another assumption is made and the curves gone through again. Having two trials, a third trial should be final since the radial stress at the bore will vary as a straight-line function with the tangential stress at the outer diameter. Usually four attempts will be made, though, since the sum and difference curves as now plotted (the same form as described by Haerle[3] is still widely used) are not sufficiently scaled to read the stress to three places. So on the third attempt the calculator usually finds that the bore radial stress is not quite zero and will go through one more trial to get a "clean" answer. Furthermore, even before the first trial the calculator often runs "off scale" and must readjust his disk speed to stay on the paper.

In the method described in this article the tangential stress at the outside is assumed (10,000 psi in the example) just as in the sum and difference curve method but, instead of using the curves, the stress condition at the bore is found by filling out the table as illustrated in Table II. The example is for the disk shown in *Fig.* 2, on which the radial stress at the periphery is 4850 psi.

First the constants and the values indicated in Table I are listed. The stress condition that exists for Table II at the bore is the bore stress for the assumed tangential stress (10,000 psi) at the outside diameter. Whatever the radial stress is at the bore, a radial stress of opposite sign but the same numerical value could be superimposed and the radial stress at the bore would become zero and the stress distribution throughout the disk would change. Table III, is then filled out, using the same Equations 7, 8, 9, and 10 except that all terms that are affected by centrifugal force are neglected. The same assumed tangential stress at the outside diameter can be used for convenience.

The superposition involving Tables II and III to secure zero radial stress at the bore can be justified

**Table IV—Final Check of Actual Stresses in Disk**

| | [N] | [P] | [Q] | [R] | [S] | [T] | [U] | [V] |
|---|---|---|---|---|---|---|---|---|
| $n$ | $p_n'$ | $q_n'$ | $p_n'+q_n'$ | $p_n'-q_n'$ | $p_n''+q_n''$ | $p_n''-q_n''$ | $p_n''$ | $q_n''$ |
| | $\frac{[U]}{[E]}$ | [V]−[F]×[N] | [N]+[P] | [N]−[P] | [Q]+[L] | [P]×[J]+[M] | $\frac{[S]-[T]}{2}$ | $\frac{[S]-[T]}{2}$ |
| 1 | 4850 | 11000 | 15850 | —6150 | 17560 | —5800 | | |
| 2 | 10800 | 13155 | 23955 | —2355 | 28145 | —500 | 5880 | 11680 |
| 3 | 8900 | 12840 | 21740 | —3940 | 25570 | —3100 | 13820 | 14320 |
| 4 | 7870 | 13325 | 21200 | —5450 | 26870 | —8780 | 11235 | 14335 |
| 5 | 6460 | 17050 | 23510 | —10590 | 25690 | —25770 | 9045 | 17825 |
| 6 | | | | | | | —40 | 25730 |
| 7 | | | | | | | | |
| 8 | | | | | | | | |

as follows. The stress condition indicated in TABLE II is due to centrifugal force and a radial pull at the bore which turns out to be 5110 psi. The conditions indicated in TABLE III are due only to a radial pressure at the bore of 51,225 psi. Then superposition of the proper part of the stresses in TABLE III on TABLE II to eliminate the radial pull of 5110 psi will involve no change in the centrifugal loading.

Summarizing what TABLE II shows it is evident that this is the stress distribution of the disk under the conditions assumed, namely a radial stress of 4850 psi and a tangential stress of 10,000 psi at the outer diameter. The radial stress 5110 psi at the bore indicates a pull of that amount is necessary to give the outside conditions. In TABLE III the stress distribution is that of a stationary disk with a radial pressure of 51,200 psi at the bore.

Having correctly filled out TABLES II and III, the bore stress can quickly be determined. To have zero radial stress at the bore in TABLE II a radial pressure of 5110 psi must be superposed on the bore pull of 5110 psi, or 5110/51,225 = 0.0998 of the radial pressure indicated at the bore in TABLE III. Then the stress distribution of TABLE II changes likewise throughout the disk and to find the correct values it is only necessary to multiply the values in TABLE III by 0.0998 and add to the values in TABLE II, taking due account of the signs. Thus the free bore stress becomes 18,630 + 0.0998 × 70,800 = 25,700 psi. It is seen that the assumed tangential stress at outer diameter in TABLE II should have been 10,000 + 0.0998 × 10,000 = 11,000 psi.

For a check on the arithmetic involved in the tables, TABLE IV can be filled out, using the new

**Nomenclature**

| | | |
|---|---|---|
| $A,B$ | = | Constants |
| $p$ | = | Radial stress |
| $q$ | = | Tangential stress |
| $r$ | = | Radius |
| $\rho$ | = | Mass density |
| $\omega$ | = | Angular velocity |
| $\mu$ | = | Poisson's ratio |
| $h$ | = | Disk thickness |
| $k$ | = | $-p/q$ for static disk at the bore |
| $\delta$ | = | Fit of disk on shaft |
| $D$ | = | Nominal disk bore diameter |
| $\triangle D$ | = | Increase of bore diameter due to centrifugal force |
| $E$ | = | Modulus of elasticity |

tangential stress at the outside of 11,000 psi. The radial stress at the bore now comes out closely zero and there is the same tangential bore stress as determined in the previous paragraph. A radial bore stress checking within plus or minus 100 psi will indicate sufficient accuracy.

The results of TABLE III are also useful in that the ratio between the radial and tangential stress is readily available for the static condition. This ratio, $k$, is necessary to find the tangential and radial bore stresses due to a given press or shrink fit, $\delta$, between the disk and the shaft. To solve for these stresses at the bore the known relations can be written:

$$\left.\begin{aligned} k&=-\frac{p}{q} \\ p&=-kq \end{aligned}\right\} \qquad (11)$$

It is convenient to inject the minus sign since $p$, the radial stress, is a compressive stress under the conditions described and $k$ then will be a positive number. Further, from Hooke's law, at the bore,

$$\frac{\delta}{D}=\frac{1}{E}(q-\mu p) \qquad (12)$$

Combining Equations 11 an 12,

$$q=\frac{\delta}{D}\times\frac{E}{1+\mu k} \qquad (13)$$

Equations 11 and 13 give the radial and tangential bore stresses due to the fit, $\delta$, for an inelastic shaft. This assumes all the fit goes into deformation of the disk. This is not quite the case since the shaft will be subjected also to radial pressure and will undergo deformation.

If the shaft is assumed not to protrude beyond the hub of the disk, the shaft deformation can be written

$$\delta_s=-\frac{Dp}{E}(1-\mu) \qquad (14)$$

The increase in bore diameter of the disk is given by

$$\delta_D=\frac{D}{E}\times q(1+\mu k) \qquad (15)$$

Then, substituting $p = -kq$ into Equation 14 and adding Equations 14 and 15, since $\delta = \delta_S + \delta_D$, and setting equal to the total shrink fit, the equation for the tangential bore stress is

$$q=\frac{\delta}{D}\times\frac{E}{1+k} \qquad (16)$$

Actually, since the shaft extends beyond the hub the correct tangential bore stress due to the shrink fit will lie somewhere in between the values given by Equations 13 and 16. After the tangential bore stress due to the shrink fit is determined, the radial bore stress is found from the relation $p = -kq$.

To illustrate the particular example shown in this article let it be assumed that the fit, $\delta$, of the disk on the shaft is 0.022-inch. For a stationary disk, and considering the shaft as rigid, the tangential bore stress, $q$ from Equation 13 is,

$$q=\frac{0.022}{8.00}\times\frac{29\times10^6}{1+0.3\times0.724}=65{,}400 \quad \text{psi}$$

Then from Equation 11 the radial pressure at the bore is

(*Continued on Page* 188)

# Reduction and Reversing Drives

***. . . for adapting gas turbines to marine propulsion. Maneuvering characteristics and economic factors are emphasized in this abstract of a recent ASME paper***


**By W. E. Hammond**

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*U. S. Maritime Commission*
*Washington, D. C.*


SINCE the characteristics of turbine type machinery provide optimum efficiency at high rotative speeds while marine propellers require low rotative speeds for optimum efficiency, it is necessary to utilize a transmission system that permits both objectives to be realized. Systems developed in connection with the application of steam turbines to marine propulsion, now commonplace, have experienced long development periods and have encountered many difficulties in arriving at their present position. It therefore appears desirable in the case of the gas turbine power plant to review the present state of the art before endeavoring to solve the problems introduced by the gas turbine. The particular problems to be faced in applying this prime mover to a ship lie in reversibility and maneuverability.

It is the purpose of this article to discuss briefly the various transmission systems available today. It is not intended that a solution be established for every gas turbine power plant application but rather that a general solution be sought which has the widest application.

Some of the requirements of a marine transmission system, not necessarily in order of importance are: First, it must be capable of reducing the revolutions from those of the prime mover to those required by the propeller. Second, it must be reliable and simple in operation, within the capabilities of the crews available. Third, transmission losses should be a minimum. Fourth, first cost and maintenance should be commensurate with those of the prime mover so that the advantages of the prime mover will not be nullified by the transmission system. Fifth, weight and space required should also be commensurate with that required of the prime movers, for the reason just given. Sixth, if the torque characteristics of the prime mover do not coincide with the requirements of the propeller for all conditions of operation, then the transmission system must be capable of suitable adjustment to meet these requirements. Seventh, if the prime mover is not reversible then the transmission must provide this feature. Eighth, should it be possible to disconnect the propeller from the prime mover accidentally while under load it must be possible to reestablish the connection immediately. If this cannot be accomplished the consequences can be extremely serious and result in totally wrecked machinery. Ninth, it is preferable that the transmission system selected for the merchant marine have wide application in order to reduce development cost, simplify manning problems, and reduce maintenance.

There are available today many arrangements of

*Fig. 1—Block diagram of machinery arrangement for a geared d-c transmission providing variable speed ratio*

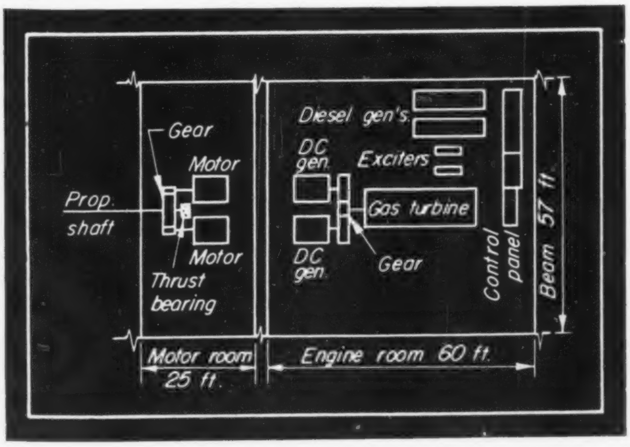
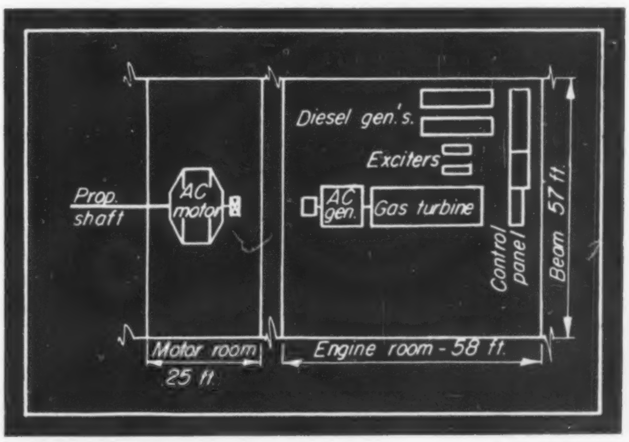



*Fig. 2—Machinery arrangement for direct-connected a-c drive which provides a fixed speed ratio*



marine transmission systems in successful operation with steam turbines, which may be applied to the gas turbine. However, there is one important difference in the two prime movers, namely, the steam turbine as it is known today incorporates a reversing unit and the gas turbine does not. True, many suggestions and proposals have been made but to date no reversible gas turbine exists.

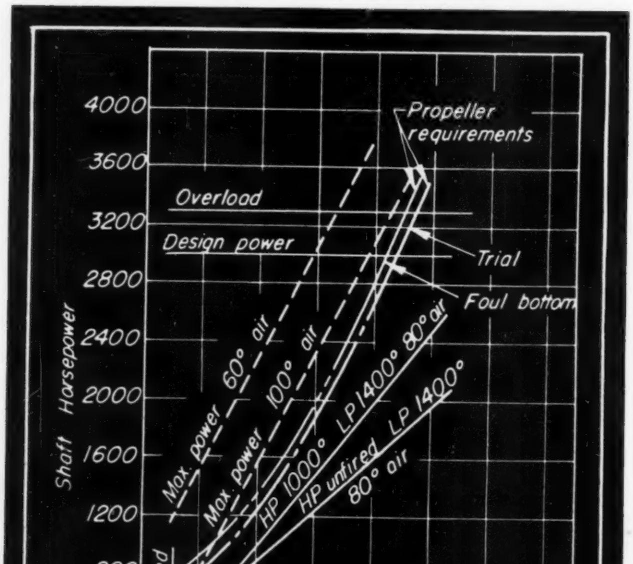
In any event, the following transmission systems presently available might be considered as possible solutions to the problem:

1. Electric drive, direct current
2. Electric drive, alternating current
3. Mechanical geared drive with controllable-pitch propeller
4. Mechanical geared drive containing reversing features
5. Mechanical geared drive with reversing gas turbine

Electric Drive, Direct Current: The d-c drive consists of one or more d-c generators connected through control gear to a d-c motor, *Fig.* 1. The generator is generally of the separately excited type to provide variable voltage control of the motor from standstill to rated speed. The motor commonly is of the compound type to provide high torque for stopping and reversing the vessel. The control gear consists of a lever for operating the forward, reverse and dynamic braking contactors and to operate the field application and generator excitation; another lever is provided to operate the prime mover governor; and a third lever for emergency stop.

This particular transmission system is in successful use in moderate numbers at the present. It has

*Fig. 3—Comparison of horsepower-speed relationships of gas turbine output and propeller requirements*




been utilized principally with diesel engines to provide rapid reversal (approximately 15 seconds from full ahead rpm to 2/3 rated rpm astern) and also to permit adjustments in propeller speed so that the torque requirements are within the engine's capacity.

Application of this drive to the gas turbine power plant does not introduce any new problems of a serious nature, provided the turbine is geared to the generator. If this is not done then either the turbine speed must be reduced or development undertaken to permit d-c generation at 4000 rpm.

In power range, the present design of this equipment is adequate for the range in shaft horsepower presently being considered for the gas turbine power plant. However, there appears to be some reason to question the application of this transmission system to large horsepowers for single units. Since a single unit is restricted to low-voltage design due to commutation difficulties, its application to large powers results in large costly equipment.

## Provides Wide Flexibility

This system is extremely flexible and will lend itself to either series or parallel connection of multiple units so that relatively large horsepowers might be accommodated by existing gas turbine designs. Furthermore, it will function equally well with any of the gas turbine arrangements whether a separate power turbine is used or not, since the torque requirement of the propeller may be adjusted to be within that of the prime mover.

It will be noted from Table I that this transmission system does not compare favorably with other systems in regard to weight, space, efficiency, and first cost. In addition, the maintenance of d-c machinery will exceed that for the mechanical drives or the a-c systems.

Electric Drive, Alternating Current: Inasmuch as the induction a-c drive has been supplanted by synchronous a-c drive, discussion will be limited to this latter type. The synchronous electric drive is essentially a constant-ratio speed-reducing transmission, as the output speed for continuous operation is always a constant ratio of the input speed. Consequently, the synchronous electric drive in itself does not provide speed or power control and is dependent upon the speed control of the prime mover to vary the speed of the propeller. The synchronous electric drive does provide means of coupling, reversing, and vibration isolation independent of the direction of rotation of the prime mover. An additional feature of the synchronous drive is its ability to provide electric power for auxiliary applications.

The drive consists of one or more synchronous a-c generators connected through control gear to a synchronous a-c motor, *Fig.* 2. The generator is provided with special field coils and exciter to permit field forcing of the generator at subnormal frequency during induction operation of the synchronous motor while starting and reversing the propeller. The salient-pole type synchronous motor is provided with heavy copper squirrel-cage windings in the pole faces to permit short time operation as an induction motor during propeller shaft starting, stopping and revers-

ing. The control gear consists of a starting lever, governor control lever, and emergency stop lever.

This transmission system has experienced wide service in the marine field, many merchant and naval vessels presently in service being so equipped. Disregarding applications made during the war for production reasons, this drive is generally applied for the purpose of obtaining high economy at low power operation and/or for applications in which it is desired to utilize the main machinery for an auxiliary purpose such as operating pumps on a tanker or dredge.

Application of this type of transmission to the gas turbine power plant should not prove difficult for those gas turbine arrangements whose torque characteristics approximate those of the steam turbine. However, for some arrangements of gas turbine power plants the torque-speed range is rather narrow and, as a result, some difficulty might be experienced in matching the two characteristics. Accordingly, the application of this transmission must depend upon the type of gas turbine power plant.

Regarding the power range of this equipment, nothing need be feared since present practice includes units well in excess of any size likely to be considered for the gas turbine power plant. Furthermore, this transmission adequately lends itself to the use of multidrive units. In other words, several small units may be coupled electrically to a given motor in order to achieve the desired output. However, it must be remembered that multidrive units require parallel operation which adds to control complications and in this respect it is not as simple as a series-connected multidrive d-c system. The a-c system when used with multiple generators does not permit the full power of the connected generators to be used when one or more generators are taken off the line.

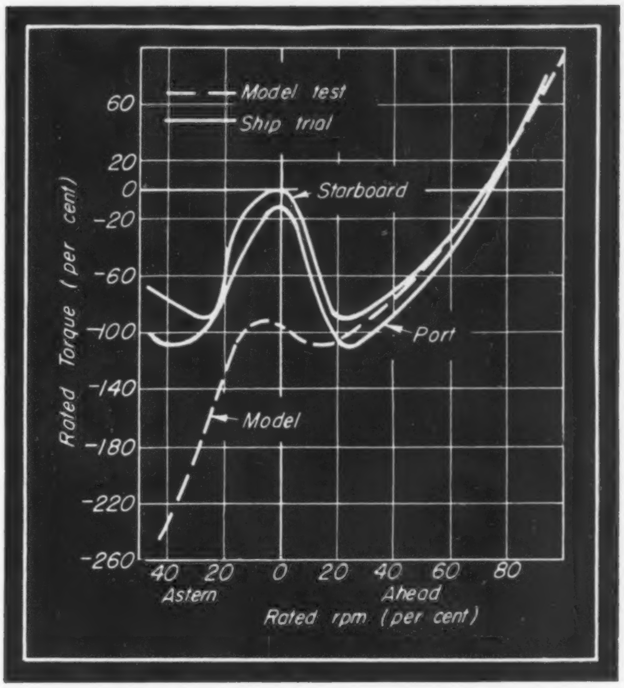



*Fig. 4—Relationships between torque and rpm during reversing of a twin-screw ship with a-c drive*

There may be a limitation in the use of this transmission system with some arrangements of gas turbine power plant. In particular, for those arrangements not containing a separate turbine for power output it is practically impossible to meet the minimum speed and torque requirement (approximately 150 per cent torque at 25 per cent speed) for maneuvering, and difficulty may be further experienced in meeting the torque requirements of the propeller,

**TABLE I—Comparison and Economic Evaluation of 6000-hp Marine Gas Turbine Drives**

| Drive[1] | Efficiency (%) | Volume (cu ft.) | Weight (lb.) | Total Cost (dollars) | Fuel Rate (lb/shp-hr.) | Fuel Cost[6] (dollars/yr) | Operating Cost[7] (dollars/yr) |
|---|---|---|---|---|---|---|---|
| Electric, AC, mfg A[2] ..... | 92.6 | 3360 | 233,200 | $240,900 | 0.486 | $147,800 | $171,890 |
| Electric, AC, mfg B[2] ..... | 90.9 | 3360 | 226,700 | 230,400 | 0.495 | 150,500 | 173,540 |
| Electric, DC, mfg A[3] ..... | 84.7 | 3640 | 320,700 | 337,000 | 0.531 | 161,500 | 195,200 |
| Electric, DC, mfg B[4] ..... | 85.3 | 3640 | 448,700 | 415,900 | 0.528 | 160,600 | 202,190 |
| Reduction gear, $K$ = 75 .. Controllable pitch propeller | 98 | 1600 | 235,000 | 309,000 | 0.459 | 139,600 | 170,500 |
| Reduction gear, $K$ = 175 . Controllable pitch propeller | 98 | 1100 | 182,000 | 241,000 | 0.459 | 139,600 | 163,700 |
| Reversing gear, $K$ = 175 .. Airflex clutch | 96 | 1600 | 135,000 | 242,400 | 0.469 | 142,800 | 167,040 |
| Reduction gear only[5] ..... For reversing turbine | 98 | 1400 | 184,000 | 209,400 | 0.459 | 139,600 | 160,540 |

[1] All drives include propellers, shafting, thrust bearings, and gears where necessary. Foundation cost not considered.
[2] Direct-connected 3600 rpm generator, direct-connected 90 rpm motor.
[3] Four geared generators and four geared motors, each running at 600 rpm. Propeller speed, 90 rpm.
[4] Geared double-armature 600 rpm generator and direct-connected double-armature 90 rpm motor.
[5] K-factor 175 in 1st reduction, 125 in 2nd reduction.
[6] Based on diesel fuel oil at $3.04 per barrel, ship at sea 60% of the time.
[7] Fuel cost plus fixed charges which are 10% of total cost of drive. Fixed charges on turbine not included.

particularly under foul-bottom conditions with high inlet temperatures, *Fig.* 3.

Insofar as arrangement is concerned, this transmission does not impose any special difficulties. However, the control equipment required is generally rather large and occupies considerable space. The weight of this transmission is appreciably less than that of d-c equipment and compares favorably with the mechanical drives, TABLE I. Considering efficiency, this system has an advantage over the d-c system but it is still appreciably below that obtained with mechanical drive. This loss of efficiency in the transmission system obviously reduces the advantages of the gas turbine application. First cost is lower than the mechanical drive with a controllable-pitch propeller, but when coupled with a lower efficiency its operating cost, including fixed charges, is about the same as the mechanical drive, TABLE I.

The operational features of the a-c transmission system involving synchronous motors need to be considered carefully. This type is rather slow in maneuvering (60 to 90 seconds from full ahead rpm to 2/3 rated rpm astern) and particular care must be taken in determining the torque characteristics of the

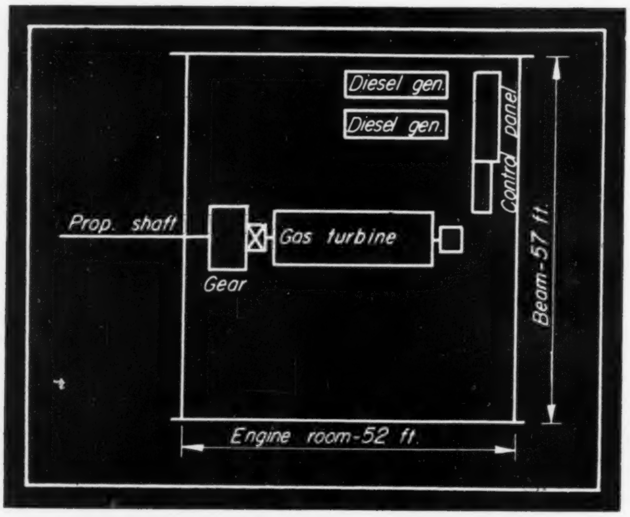



***Fig. 5—Block diagram of machinery arrangement using a reduction gear and controllable-pitch propeller***

propeller for the particular ship under consideration. In *Fig.* 4 is indicated the variation between the predicted value and the measured value of the torque required to reverse a particular vessel. Further, speed control below 20 per cent is generally not possible with this system. Another factor that must be evaluated is the maintenance of the controls. It is extremely difficult to obtain ship crews that are qualified in the operation and maintenance of this equipment so that high maintenance may result.

REDUCTION GEARS WITH A CONTROLLABLE AND REVERSIBLE-PITCH PROPELLER: This transmission system consists of the usual marine mechanical reduction gear, presently used with steam turbines, coupled with a controllable and reversible-pitch propeller. The primary function of the controllable-pitch feature is to reverse the vessel although it will also provide infinite speed variation from zero to maximum revolutions ahead and astern. This permits adjustment of the torque requirements of the propeller to match the characteristic provided by the machinery for any and all conditions that are encountered in service.

One type of controllable and reversible-pitch propeller rotates the blades through a crank arrangement in the hub. The cranks are operated through a piston rod and crosshead by a servomotor.

While the controllable and reversible-pitch propeller is a comparative newcomer to the marine field in this country, there is ample background upon which its performance and reliability may be judged. A number of installations have been made in Europe; further, a large number of Kaplan type water turbines, which for all practical purposes duplicate the function of the controllable-pitch propeller, have been used both in this country and abroad. While the largest marine propeller of the controllable and reversible design that has been built to date is 14 ft 9 in. diameter and 3500 hp, there is every reason to believe that insofar as merchant designs are concerned the limitation on diameter and power will be the same as with fixed-pitch propellers. The basis for this statement lies in the fact that the Kaplan turbines have been in successful operation in much larger sizes than would be found in marine design.

While the use of this system does not lend itself to the use of large numbers of units, it can be used with multiple drive units not exceeding four in number for each shaft. Furthermore, this system of transmission is suitable for any gas turbine arrangement proposed so far, i.e., whether a separate power turbine is provided or whether the power turbine also furnishes the compressor power.

## Layout Similar to Steam Turbine

In regard to arrangement, this system is practically identical with the mechanical geared system used with steam turbines, *Fig.* 5. However, the controllable and reversible propeller, with its shafting, is heavier than a fixed-pitch propeller and its shafting, so that in the higher horsepower ranges an increase in weight must be accepted.

The controllable-pitch propeller permits the torque characteristics to be adjusted to meet the capabilities of the prime mover for any and all conditions of service operation. This is particularly advantageous with the gas turbine should it become necessary to reduce the operating temperature in service, *Fig.* 3, or it may be used to compensate for reduction in power due to increases in inlet temperature. In this respect, it is comparable in flexibility to the d-c electric transmission system.

While the first cost of this system is higher than that of a-c electric drives owing to higher propeller cost, its higher efficiency offsets this so that the operating cost per year is one of the lowest of all drives considered, TABLE I. Further, the cost of the controllable and reversible-pitch propeller is based upon rather limited production and application. There is every reason to expect a reduction in the cost of this item as its use increases, which will further im-

(*Concluded on Page* 180)

# MACHINE *Editorial* DESIGN

## Design Now for Cost Reduction

*Though comparisons may be odious, there is one in particular that the designer should keep closely in mind due to the current trend in economic conditions. Whereas during the war the engineer's sole aim was the development of equipment which—in many cases regardless of cost—would accomplish its purpose, the picture now has changed to the point where proper consideration of all cost factors may well have a decided effect on the future stability of the nation.*

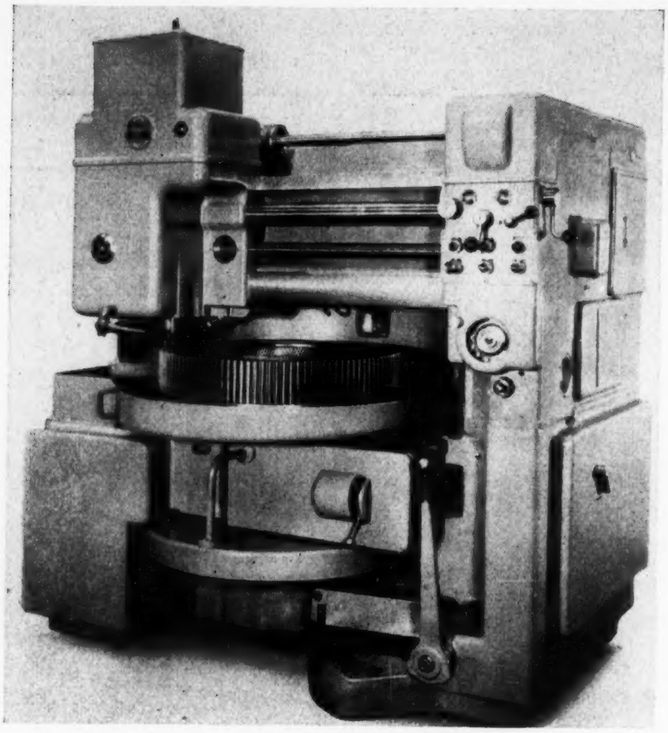
*Here is where the designer again has the opportunity to fill a role of vital importance. If he can assist his company, through foresight in design, to build an enviable reputation for performance and at the same time enable the organization to continue to market its machines at reasonable prices, he will have achieved another outstanding accomplishment.*

*Indicative of the thinking of the buying public on the matter of prices are the results of a survey presented at the recent annual meeting of the Society of Automotive Engineers. It was shown that if the major producers of automobiles were able to develop a light car to sell in the $750-$1000 price range, seventy-two per cent of the persons approached would buy one of these in preference to one of the current "lower-priced" models. In other words, it seems only a question of time—and the fulfillment of present demand—before strictly economical, down-to-earth design will become the order of the day.*

*To accomplish this—not only with regard to automobiles but all other types of machines—the designer is faced with the problem of effecting the maximum amount of redesign from the economy standpoint or, as individual cases may warrant, the development of entirely new machines in the lower price brackets. The answer is not simple, but that great strides can be made in this direction is being proved constantly. Some pertinent examples of designing for economical production are given, for instance, in the leading article in this issue of MACHINE DESIGN. Application of basic design principles such as those discussed by the author should go far in aiding the designer to meet the requirements of the future.*

*It is not enough to design only for technical supremacy; the cost involved, in relation to potential market conditions, necessarily must be taken into account.*

L. E. Jermy



# DESIGNS OF THE MONTH

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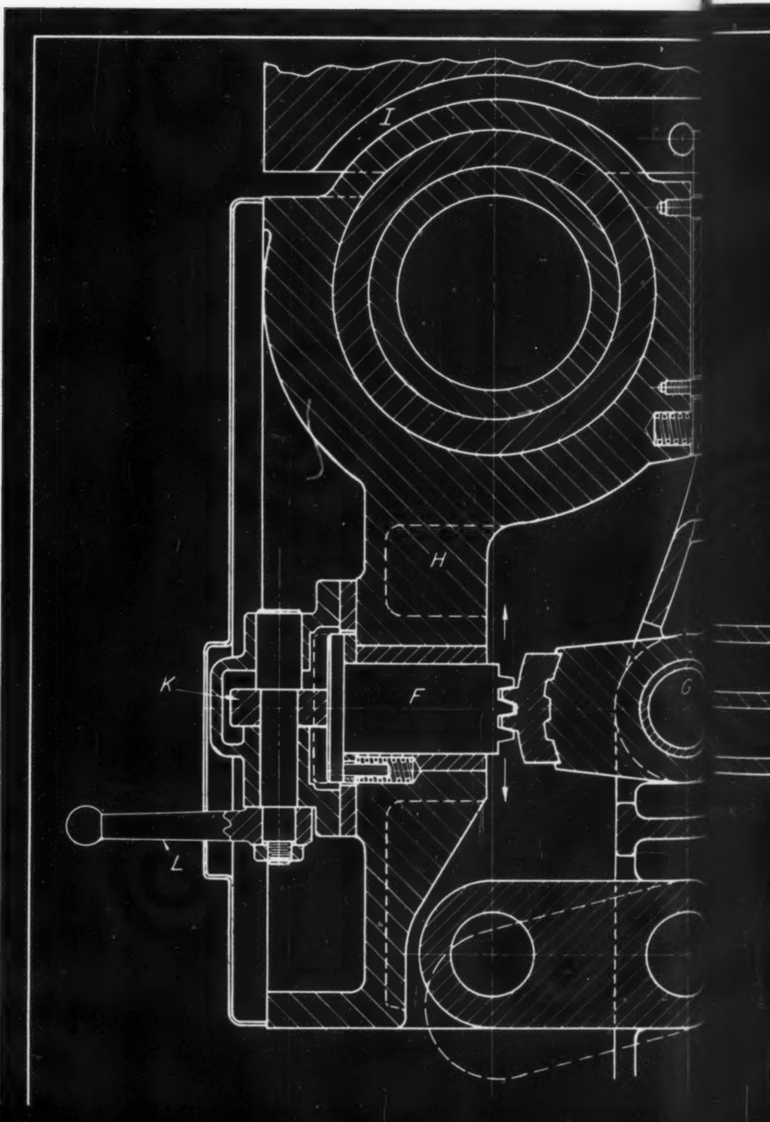
## *New Apron Relief in Gear Shaper*

DESIGNED for heavy-duty general-purpose gear cutting, the 36-inch gear shaper, above, incorporates an apron-relief mechanism unusual in its field. As the cross-section drawing, right, shows, cam *A* is mounted on a vertical shaft which makes one revolution for each cutting stroke of the cutter, and contacts rollers *B* and *C* mounted in the carrier *D* attached to the relief lever *E*. A segment on the relief lever meshes with a rack, *F*, fastened to the apron. As lever *E* oscillates on pivot *G*, it causes a movement of the work-holding apron *H*. On the cutting stroke, the apron is held into a "four-point" bearing *I*, by the high lobe on the cam-contacting roller *B*. After the cutter has passed through the work, the low point on the cam approaches roller *B*, the high lobe contacting roller *C* and thus moving lever *E*, which withdraws the work from the cutter on its return stroke.

Variously shaped cams *A* can be interchanged readily to provide different degrees of relief as conditions require. Correct alignment of the cams is obtained by aligning graduations on them with a pointer when the cutter is at the top of its stroke. Initial tension on the relieving lever is obtained by adjusting roll-carrier *D* by means of screws *J*. A slight amount of tension is set up in the lever when the apron is

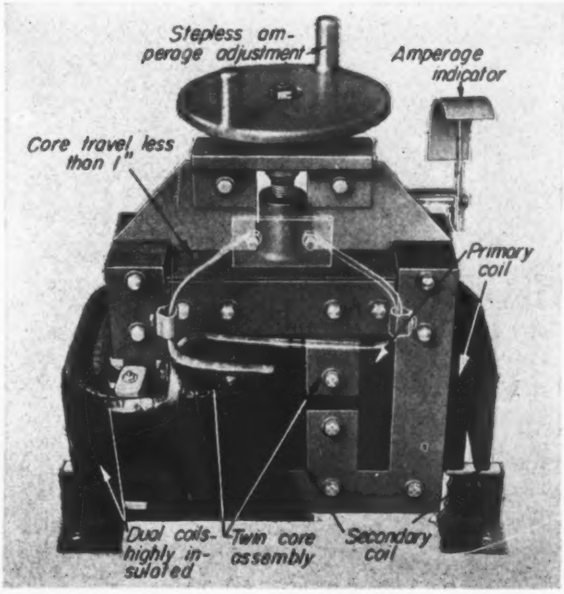


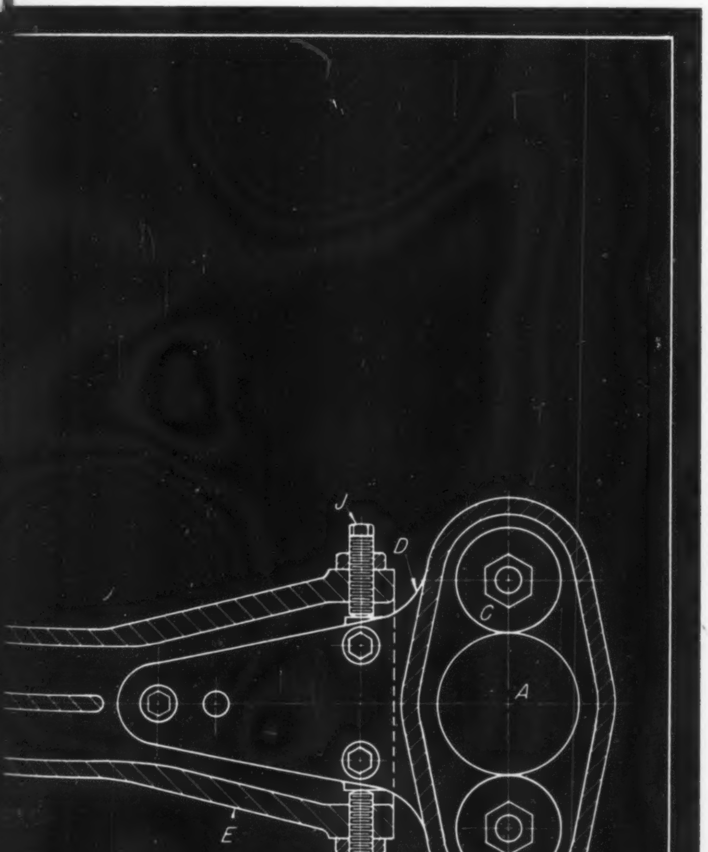
seated in the "four-point" to insure that the apron is seated properly during the cutting stroke.

When cutting large-diameter gears, it is necessary to open the apron and swing it away from the machine. This is accomplished by rotating cam *K* with lever *L*, thus disengaging the rack, which is spring loaded, from the segment on the relieving lever. This permits the operator to swing the apron by means of a lever away from the machine. Mfr.: The Fellows Gear Shaper Co., Springfield, Vt.

## Arc Welder Employs New Magnetic Circuit Design

WITHOUT use of arc boosters, reactor coils or fans, the welder, upper right, achieves stepless amperage control through a novel arrangement of dual primary and secondary coils and movable core in its transformer. As the cutaway view, right, shows, primary and secondary windings of the transformer are each divided into two coils which may be connected in series or parallel, depending

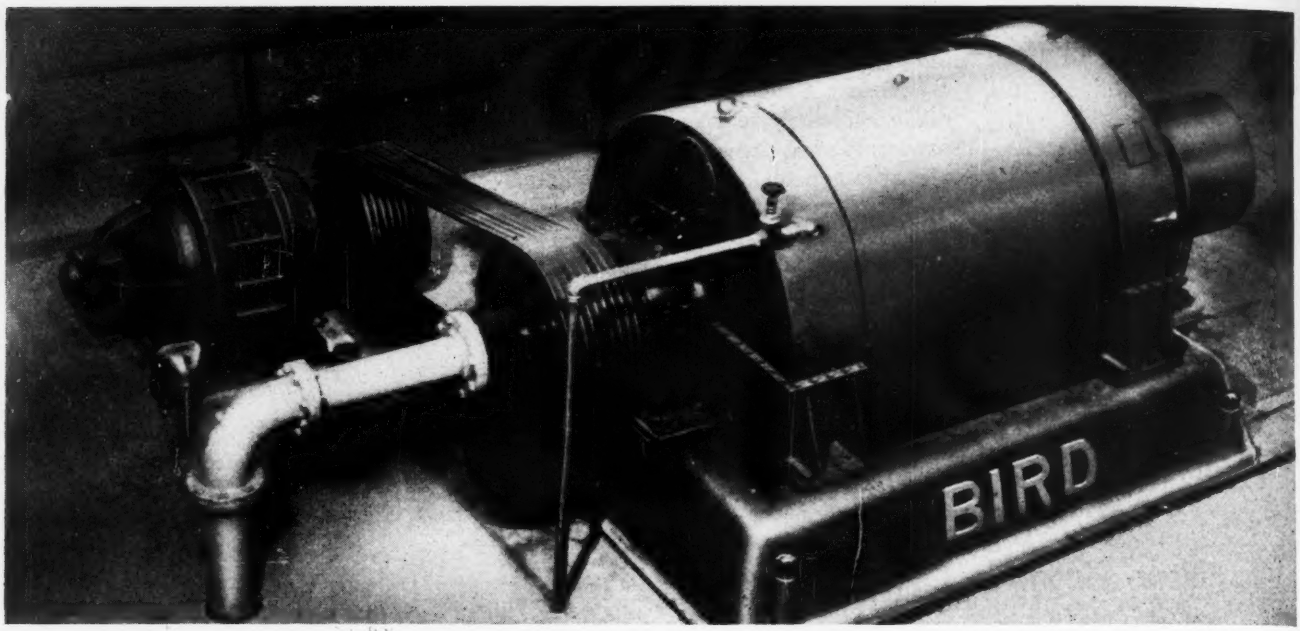



Plan Section Through Apron Rack

upon the primary and secondary voltages. In the primary magnetic circuit there is a stationary U-shaped laminated core which has two inwardly projecting secondary cores or poles on its central leg. Completion of the primary magnetic circuit is effected by a movable core —a horizontal member which slides up and down between the ends of the U. The two primary coils are wound on the two legs of the primary core.

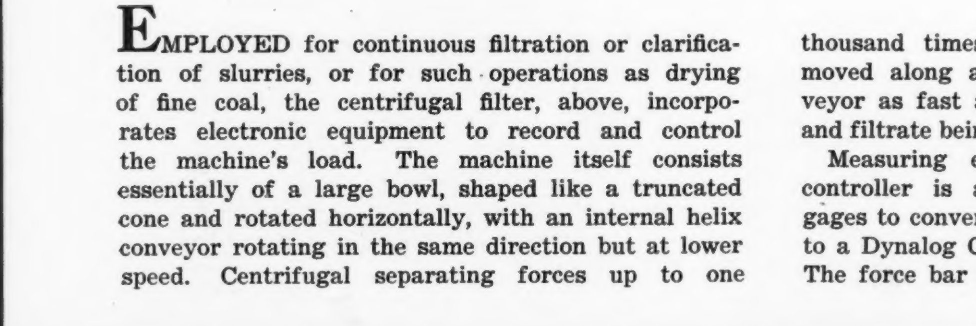
In axial alignment with the secondary poles of the stationary core are two projecting poles on the movable core. As the cutaway view shows, the secondary coils are wound around these secondary poles of both cores as well as around the primary coils. Movement of the movable core by means of the handwheel does not materially change the reluctance of the primary magnetic circuit. However, it varies the reluctance of the magnetic shunt between wide limits and thus varies the impedance of the secondary coil to give a stepless amperage control. Mfr: John A. Kern Co., Chicago 7.




## *Strain Gages Used in Recording and Controlling Machine Load*

EMPLOYED for continuous filtration or clarification of slurries, or for such operations as drying of fine coal, the centrifugal filter, above, incorporates electronic equipment to record and control the machine's load. The machine itself consists essentially of a large bowl, shaped like a truncated cone and rotated horizontally, with an internal helix conveyor rotating in the same direction but at lower speed. Centrifugal separating forces up to one thousand times gravity are developed. Solids are moved along and out of the machine by the conveyor as fast as deposited, discharge of both solids and filtrate being continuous.

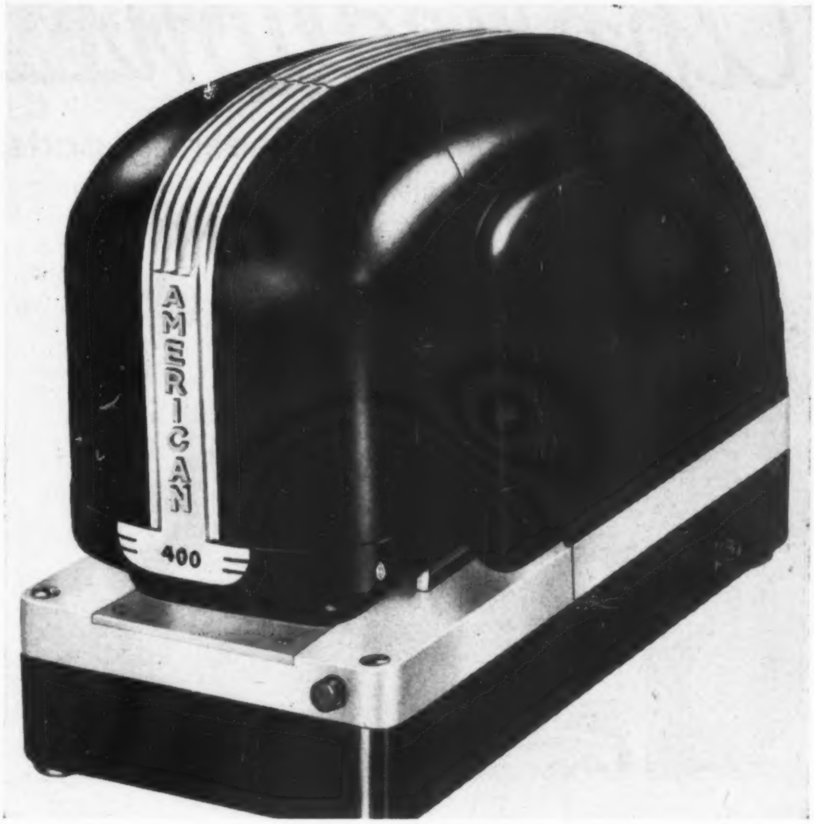
Measuring element of the electronic recorder-controller is a calibrated force bar using strain gages to convert force changes which are transmitted to a Dynalog Controller (see schematic, left, below). The force bar is operated by a torque arm fastened to the stationary gear of the planetary gear system used to transmit motion from the bowl to the conveyor. As the machine load increases, the force on this stationary gear increases, increasing the force on the measuring element and recording this force on the instrument. The instrument chart is graduated in pounds of force and readings may be converted to machine load by a factor appropriate to the installation.



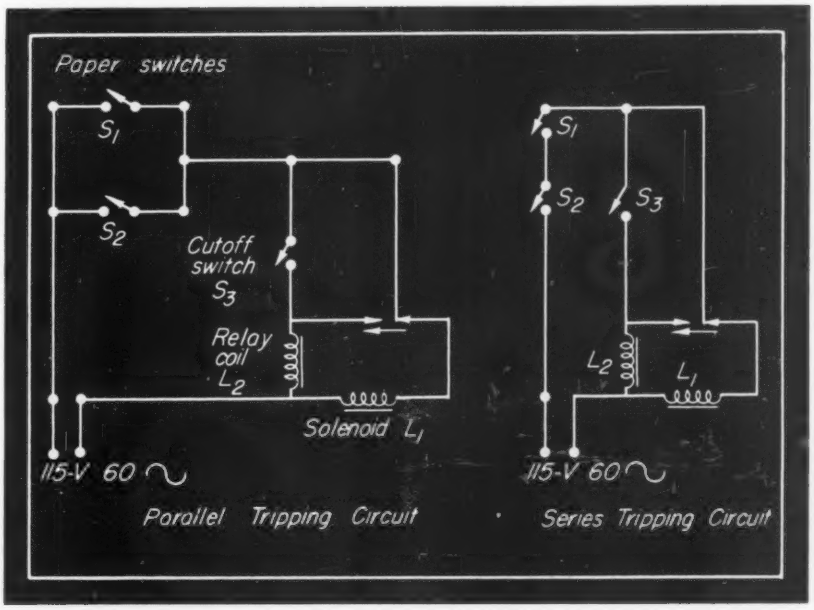

To maintain the load at a desired point for maximum efficiency, pneumatic control generally is used to position a throttling device in the slurry feed line with electric control usually added for safety. A typical safety control functions to first shut down the slurry feed if the machine loading tends to rise too high due to overfeeding. If this does not

permit the condition to clear itself and the loading continues to rise, a second circuit shuts off the power, stopping the machine. A third circuit shuts off the power in the event the safety pin connecting the differential gear drive is sheared or the machine runs free with no force on the bar. Mfr. of machine: Bird Machine Co., South Walpole, Mass. Mfr. of recorder-controller: The Foxboro Co., Foxboro, Mass.

## Fool-Proof Tripping Circuits in Electric Check Perforator




INSERTION of paper edge against the extension arms of sensitive switches actuates the portable electric check cancelling perforator, right. As the diagram, lower right, shows, provision is made for connecting the two "paper switches" either in parallel or series. Series connection will enforce a perfectly level and complete perforation with uniform registration of the perforation no matter how the operator handles the items. Parallel connections, which will cause the machine to actuate the moment either one of the two paper switches is closed, will allow a careless operator to insert some items crookedly so that only part of the perforated marking will strike them, but will be slightly faster than operation with the series circuit.

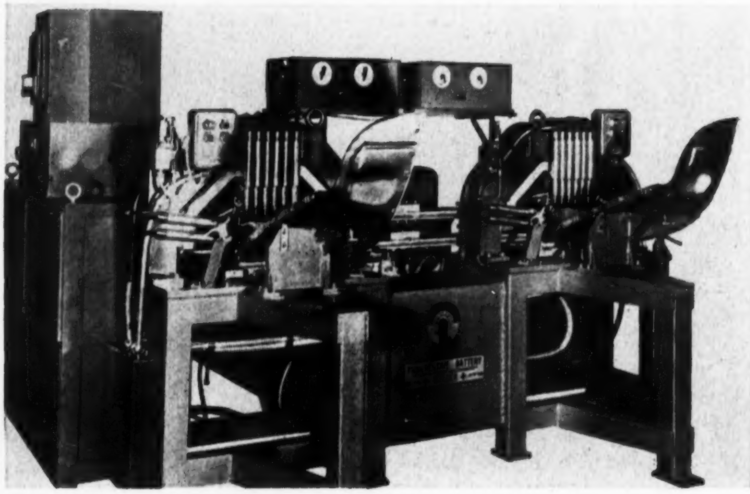



When the circuit through the two paper switches $S_1$ and $S_2$ is closed, solenoid $L_1$ is energized, closing the motor-switch contact points and engaging the clutch. The contact points can be so adjusted that they close a fraction of a second before the clutch engages and this allows the motor to build up sufficient torque. As soon as the mechanism starts its cycle, the auxiliary cutoff switch $S_3$ is closed, thus energizing coil $L_2$ of the relay. This immediately locks the solenoid out of the circuit and holds the relay energized, thus continuing to lock the solenoid out of the circuit until removal of the item that has been perforated opens the paper switches.

At the end of the cycle, a single-revolution clutch automatically disengages and opens the motor switch points. Thus, if the perforated item is left in the die block, there is no danger of its being mutilated by a second perforation because the relay will still be energized and the solenoid will still be locked out of the circuit. The moment the paper switches are opened, the relay returns to its normal position and the mechanism is ready for another cycle. Mfr.: The American Perforator Co., Chicago 6.

# Applications

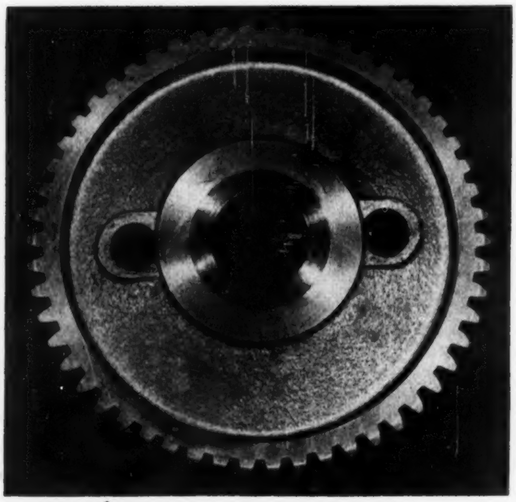
## of engineering parts, materials and processes



### Welds in Two Shots

UP TO 300 dust shields are fabricated each hour by the Progressive stored-energy multiple-spot welder, left. In this double-station machine eleven welds are completed in two "shots", the first producing six simultaneous welds and the second producing the other five. The electrodes also function as clamps for holding work in the machine. With previous spot-welding systems the same crew was able to produce only 125 units an hour.

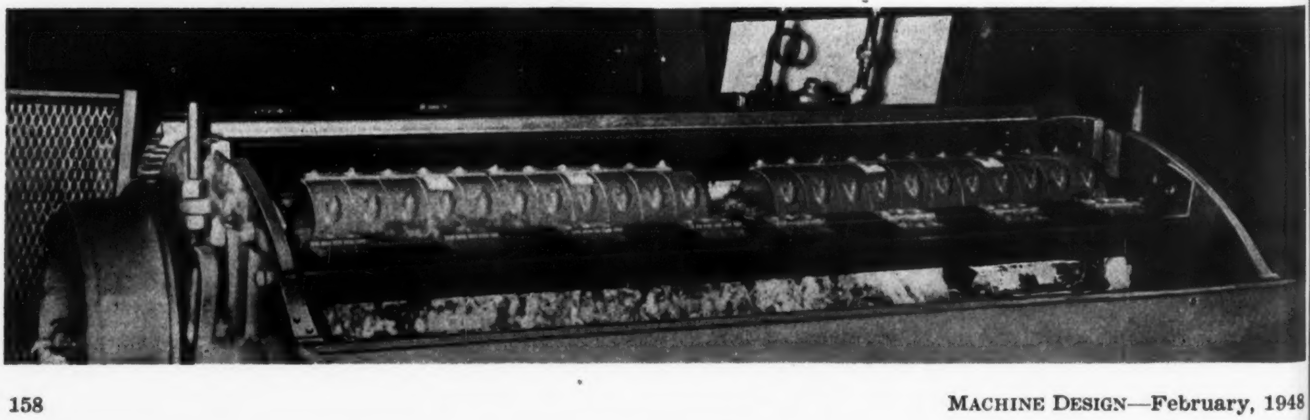
### Bond Increases Gear Strength



CHEMICAL bonding between steel bushing and aluminum body of timing gear, left, has tensile strength of 6000 psi. As a result, gear will not slip on its hub even under unusually heavy loads. Under breakdown test, gear similar to one shown was run 123 hours at 4000 rpm, followed by over 86,000 miles of truck operation with no appreciable wear. This bonding technique known as the "Al-Fin" process allows advantage to be taken of the low friction coefficient of aluminum for the gear and the strength and fatigue resistance of steel for the hub.

### Reduces Fire Hazard

DANGER of fire, caused by sparks from tramp iron in wool picker, below, has been largely reduced by the use of Eriez permanent magnets, shown below cover. Magnets remove ferrous material without interfering with picker operation. Unit is attached to frame of picker and is hinged for cleaning.



# Cam Curvature Formulas

### . . . aid calculation of instantaneous profile radius when motion of the follower is known

By M. L. Baxter Jr.
*Gleason Works*
*Rochester, N. Y.*

ONE of the more common problems facing a machine designer is that of determining the smallest cam capable of producing a prescribed motion or, conversely, determining what extremes of motion can be obtained from a cam of limited size. Two criteria provide the basis for the solution of this problem—pressure angle and cam curvature.

Pressure angle is the angle between the direction of force and the direction of instantaneous motion. It governs the amount of side thrust on the follower guides (and on the cam milling cutter in manufacture), and depends primarily on the instantaneous follower velocity.

Importance of curvature is illustrated in *Fig.* 1, which shows a circular follower riding on a cam. If it

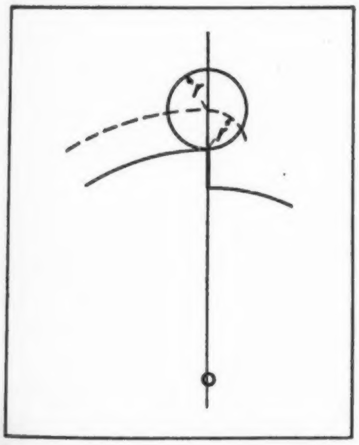


*Fig. 1—Showing how use of a roller follower results in limiting the maximum rate of deceleration*

is desired to move the follower toward the cam center as rapidly as possible, starting at the position shown, it is apparent that the best that can be done is to cut the cam curve off sharply. The center of the follower, however, does not immediately move toward the cam center; it proceeds about a circular path representing a finite limiting deceleration for the assumed dimensions. If a greater deceleration is required, either the cam size must be increased or the follower size decreased. Therefore, a primary rule of cam design is that the required motion must not produce a radius of curvature of the pitch curve less than the radius of the follower. The amount by which this limit must be avoided in practice depends on the allowable compressive stresses in cam and follower.

This data sheet presents a method of determining the cam curvature, as well as pressure angle, for a given set of dimensions and for all cases of follower motion:

1. Circular follower, radial translation
2. Circular follower, offset translation
3. Circular follower, rotation
4. Flat follower, translation
5. Flat follower, rotation

The corresponding formulas are listed in TABLE I. It will be seen that the formulas assume a simplified form for the special case where the follower velocity is momentarily zero. In many instances this is also the point of maximum curvature, as discussed later in this data sheet.

CHARACTERISTICS OF FOLLOWER MOTION: To the order required for the determination of cam curvature, the follower motion is sufficiently defined by the quantities of $x$ and $y$ for translation, and $\chi$ and $\psi$ for rotation. While it is unnecessary to bring in the concept of time, these quantities may be related to actual velocity and acceleration by the following equations:

$$x=\frac{v}{\omega_c} \qquad (1)$$

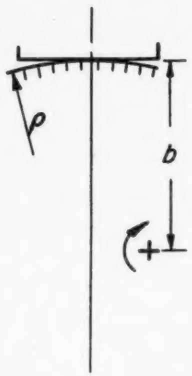
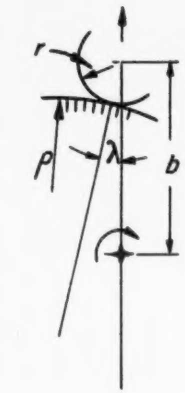
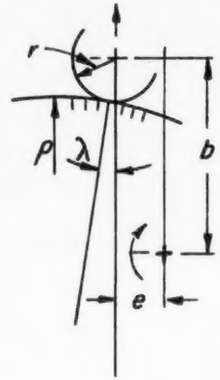
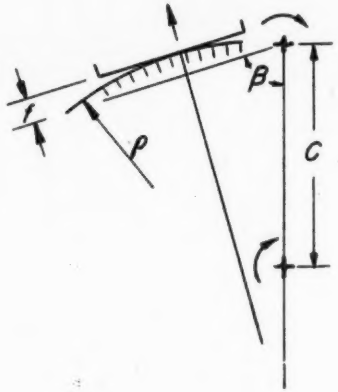
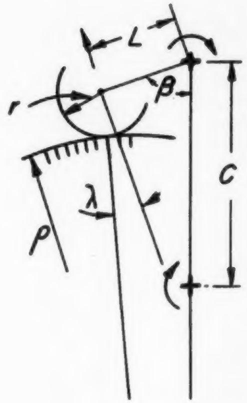
$$y=\frac{a}{\omega_c^2} \qquad (2)$$

$$\chi=\frac{\omega}{\omega_c} \qquad (3)$$

$$\psi=\frac{\alpha}{\omega_c^2} \qquad (4)$$

This data sheet is based on a paper presented before the Machine Design division of ASME at the society's recent annual meeting in Atlantic City.

**TABLE I—Pressure Angle and Cam Curvature Formulas**

| | Translation | | | Rotation | |
|---|---|---|---|---|---|
| | **Flat Follower** | **Circular Follower Radial Motion** | **Circular Follower Offset Motion** | **Flat Follower** | **Circular Follower** |
| |  |  |  |  |  |
| **Velocity = 0, High or Low Point** | | | | | |
| **Pressure Angle** | | $\lambda=0$ | $\tan\lambda=\frac{-e}{b}$ | | $\tan\lambda=\operatorname{ctn}\beta-\frac{L}{C\sin\beta}$ |
| **Curvature** | $\rho=b+y$ | $\frac{1}{\rho'}=\frac{1}{b}\left(1-\frac{y}{b}\right)$ $\rho=\rho'-r$ | $\frac{1}{\rho'}=\frac{\cos\lambda}{b}\left(1-\frac{y}{b}\cos^2\lambda\right)$ $\rho=\rho'-r$ | $\rho''=C[\sin\beta+\psi\cos\beta]$ $\rho=\rho''+f$ | $\frac{1}{\rho'}=\frac{\cos\lambda}{C\sin\beta}\left[1-\frac{L\cos^2\lambda}{C\sin\beta}\psi\right]$ $\rho=\rho'-r$ |
| **Velocity ≠ 0, General Case** | | | | | |
| **Pressure Angle** | | $\tan\lambda=\frac{x}{b}$ | $\tan\lambda=\frac{x-e}{b}$ | | $\tan\lambda=\operatorname{ctn}\beta-\frac{L(1-\chi)}{C\sin\beta}$ |
| **Curvature** | $\rho=b+y$ | $\frac{1}{\rho'}=\frac{\cos\lambda}{b}\left[1+\frac{\cos\lambda}{b}(x\sin\lambda-y\cos\lambda)\right]$ $\rho=\rho'-r$ | | $\rho''=\frac{C}{(1-\chi)^2}\left[(1-2\chi)\sin\beta+\frac{\psi}{(1-\chi)}\cos\beta\right]$ $\rho=\rho''+f$ | $\frac{1}{\rho'}=\frac{\cos\lambda}{C\sin\beta}\left\{1+\frac{L\cos\lambda}{C\sin\beta}\left[\chi(1-\chi)\sin\lambda-\psi\cos\lambda\right]\right\}$ $\rho=\rho'-r$ |

Where $v$ = follower velocity, translation, in. per sec $= db/dt$; $a$ = follower acceleration, translation, in. per sec$^2$ $= dv/dt = d^2b/dt^2$; $\omega$ = follower velocity, rotation, rad per sec $= d\beta/dt$; and $\alpha$ = follower acceleration, rotation, rad per sec$^2$ $= d\omega/dt = d^2\beta/dt^2$.

Where the follower motion is readily defined mathematically, as for constant acceleration or harmonic motion, $x$ and $y$ are easily obtained from this knowledge, using Equations 1, 2, 3, or 4. Otherwise they may be easily obtained from a table of successive follower positions such as is often required for cam manufacture, using the relations:

$$x = \frac{180}{\pi}\frac{d_1}{\Delta\theta} \qquad (5)$$

$$y = \left(\frac{180}{\pi}\right)^2 \frac{d_1}{(\Delta\theta)^2} \qquad (6)$$

$$\chi = \frac{d_1}{\Delta\theta} \qquad (7)$$

### Nomenclature for Table I

$\rho$ = Radius of curvature of cam at point of contact

$r$ = Radius of follower for a circular follower

$\rho'$ = Radius of curvature of path of follower center for a circular follower $= \rho + r$

$\lambda$ = Pressure angle = Angle between the direction of force and the direction of instantaneous motion

$\theta$ = Angle of cam rotation

*Translational Motion*

$b$ = Distance from cam center to center of follower, for a circular follower, in direction of motion

= Distance from cam center to follower surface for a flat follower

$e$ = Offset of follower path from cam center for a circular follower

$x = db/d\theta$ = Distance proportional to instantaneous follower velocity

$y = dx/d\theta = d^2b/d\theta^2$ = Distance proportional to instantaneous follower acceleration

*Rotational Motion*

$\rho''$ = Radius of curvature of cam where follower surface contains turning center of follower, for flat follower

$C$ = Center distance between turning center of cam and turning center of follower

$L$ = Distance from turning center of follower to center of follower, for a circular follower

$f$ = Offset of follower surface from turning center of follower, for a flat follower

$\beta$ = Angle between line of centers and center line of follower, for a circular follower

$\chi = d\beta/d\theta$ = Quantity proportional to instantaneous follower velocity

$\psi = d\chi/d\theta = d^2\beta/d\theta^2$ = Quantity proportional to instantaneous follower acceleration

# Engineering Data Sheet

$$\psi = \frac{180}{\pi}\frac{d_2}{(\Delta\theta)^2} \qquad (8)$$

Where $\Delta\theta$ = tabular interval of cam rotation, degrees; $d_1$ = average of first differences on each side of assumed tabular value, inches or degrees; $d_2$ = second difference at assumed tabular value, inches or degrees.

Point of Maximum Curvature: It is possible, by differentiating the appropriate curvature formula, to set up a relation between the variables for the condition of maximum curvature. In the general case such a relation is too complicated to be of practical use, and it is far shorter to calculate the curvature at several points and to plot these results to determine the worst position.

An analytical determination of the point of maximum curvature obviously requires an analytic expression for the follower motion. Considering two of the most common follower motions, harmonic and constant-acceleration, considerable information as to the location of the critical point may be obtained for the simpler cases.

Considering only translational follower motion, a portion of the cam corresponding to rotation of $\theta_R$ radians may be studied. At both ends of this interval the follower velocity is zero, the follower distance decreasing during the rotation from $b_x$ to $b_n$. Under these conditions, the following rules apply for the location of the point of maximum curvature:

*Flat Follower*: The maximum curvature always occurs where $b = b_x$, for both harmonic and constant-acceleration returns. In the special case of harmonic motion where $\theta_R = 180°$, the curvature is constant and the cam profile is a semicircle, representing the well-known case of the eccentric circle.

*Circular Follower—Constant Acceleration*: Maximum curvature occurs for $b = b_x$ when

$$\left(\frac{b_x - b_n}{b_x}\right) > 0.316\,\theta_R^2 \qquad (9)$$

This represents the majority of cases where danger of a cusp exists; only where a relatively large roller is used will it be necessary to investigate other points.

*Circular Follower—Harmonic Motion*: Here again the maximum curvature usually occurs when $b = b_x$. The necessary condition is

$$\left(\frac{b_x - b_n}{b_x}\right) > \left(\frac{\theta_R}{\pi}\right)^2 \left[1 + \sqrt{\frac{7}{3} - \frac{4}{3}\left(\frac{\pi}{\theta_R}\right)^2}\,\right] \qquad (10)$$

### References


1. *Machine Design*—Berard and Waters, 2nd edition, D. Van Nostrand Co. Inc., New York.
2. "Cam Design and Analysis for High Speeds and Torques"—Milton A. Sanders, *Product Engineering*, March, 1947.
3. "Kinematics of Disk Cam and Flat Follower"—Allan H. Candee, *ASME Transactions*, Vol. 69, No. 7, Oct., 1947.

# Assets to a Bookcase

### The Theory of Mathematical Machines

*By Francis J. Murray, associate professor of mathematics, Columbia University; published by King's Crown Press, Morningside Heights, N. Y.; 118 pages, 8½ by 11 inches, paperbound; available through* MACHINE DESIGN, *$3.00 postpaid.*

Constituting one of the first efforts to present a comprehensive discussion of the basic principles of calculating machines, this volume covers the theory and mechanics of computors from simple counters to harmonic analyzers. While concerned with the mathematical approach to design, the book is essentially an engineering text, for in virtually all instances discussion centers around practical mechanical or electrical devices for duplicating mathematical operations. The book is well illustrated, drawings being resorted to in most instances to assist the description. Contents include: Discussion of digital machines such as counters and the punch-card machine; continuous operators such as adders, multipliers and integrators; problem solution; and mathematical instruments including the planimeter, and the harmonic analyzer.

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### SPI Handbook

*Published by the Society of the Plastic Industry Inc., New York; 451 pages, 6 by 9 inches clothbound; available through* MACHINE DESIGN, *$7.50 postpaid.*

The product of the labor of over 300 engineers and more than 600 engineering firms, this mammoth work was over six years in preparation. It can be said conservatively that the result is worthy of their efforts. Those plastic materials of importance from the industrial viewpoint have been most carefully appraised, discussed and tabulated and the various aspects of design and manufacture treated in detail.

Chapters one, three, four, five, six and ten are of particular interest to the designer. Chapter one is a discussion of the SPI classification tables covering in detail the meaning of various properties and the use of classification codes. Chapters three and four, covering design, treats such aspects as radii, fillets, tapers, bosses and inserts; 66 pages being devoted to these subjects. Standards for tolerances on molded parts are discussed in Chapter five. Twenty tables are presented, covering phenolics, urea and melamine resins, cellulosics, methacrylates, and polystyrene. Problems in cementing and assembly of these and other plastics are covered in Chapter six. Laminated products with sizes and properties presented largely in tabular form, are the subject of Chapter ten. The balance of the book is devoted to such aspects as molding, forming, machining and testing.

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### Practical Design Handbook for Engineers

*By Dr. Alois Cibulka, consulting engineer; published by Clarke & Courts, Houston, Texas; 400 pages, 9½ by 12 inches, paperbound; available through* MACHINE DESIGN, *$6.00 postpaid.*

A reference book rather than a text, this volume presents in tabular and chart form a wealth of data in the mechanical, hydraulic and structural fields. In a concise manner it goes over the fundamentals of the subject, presents a typical problem worked out in detail and follows up with unusually complete tables of the data required for calculations. Coverage includes structures with much data and many shear diagrams of statically-indeterminate types, pressure vessels, shafts, gearing, and hydraulics. The last portion of the book is devoted to useful mathematical and conversion tables.

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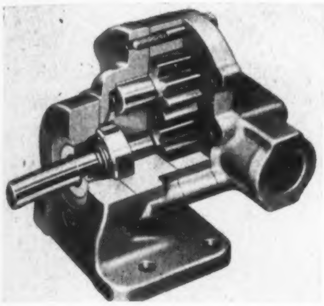
Useful discussion of decorative and protective finishes, published under the title "Finishes for Aluminum" is contained in a 5¾ by 8¾-inch paperbound, 108-page volume, which is to be supplemented at a later date by a 120-page second section containing shop data. Such aspects as mechanical surface finishes, chemical finishes, electroplating, painting, ceramic coating and silk-screening are discussed, together with controls and methods of testing finishes. Copies are available at $2.00 from Reynolds Metals Co., Dept. 27, 2500 S. Third St., Louisville 1.

"A Bibliography on Die Casting" is a careful index of books and articles relating to that general subject. Over 1200 references are included, broken down into the categories: Alloys, applications, design, finishing, machines, processes and properties. Indexing is by authors rather than titles. The 8½ by 11-inch paperbound volume is 74 pages in length and priced at $7.50. It is available from The Technical Publishing Co., 1240 Ontario St., Cleveland 13.

# new parts and materials

To obtain additional information on these new developments see Page 279

## Gear-Type Hydraulic Pumps



Medium - pressure pumps are made in models rated 1.5 to 60 gpm at 1800 rpm and 1000 psi. The pumps are available in foot, flange or other mountings. Features of the new pumps include special leakproof seal, eliminating packing difficulties, and a ball-type shaft key which permits the pump gears to float laterally. This novel keying system is said to prevent thrust loads from being transferred from the shaft to the gears or end plates and insure perfect alignment of the gears at all times. Manufacturer: Adel Precision Products Corp., Burbank, Calif.

For further information circle MD 1 on card Page 279

## Heavy-Metal Alloy

Tungsten-copper-nickel alloy having density fifty per cent greater than lead is recommended for designs requiring great inertia and small size. The material, known as Hevimet, is sintered and has density of 16.8 to 17 gm per cu cm. Tensile strength is 85,000 to 118,000 psi and hardness is 30 to 40 Rockwell C. Manufacturer: General Electric Co., Pittsfield, Mass.

For further information circle MD 2 on card Page 279

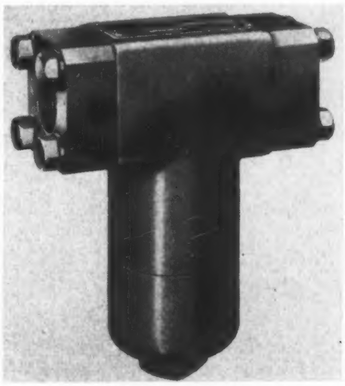
## Flexible Coupling

Recent design improvement in the American Flexible coupling is said to increase the wearability as much as 30 per cent. Wear-absorbing part of this

Oldham-type coupling is the sliding center member. This is faced with bearing strips which are surfaces of contact with the jaw flanges. New improvement allows bearing strips to slide upon the center block to which they are secured. This is said to reduce greatly the friction which occurs between the strips and the flanges as the coupling rotates. Manufacturer: American Flexible Coupling Div., J. A. Zurn Mfg. Co., 1801 Pittsburgh Ave., Erie, Pa.

For further information circle MD 3 on card Page 279

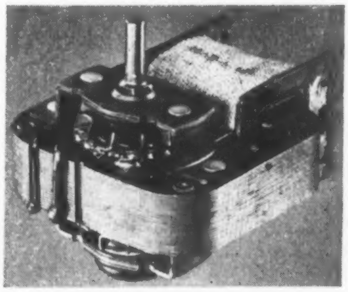
## Hydraulic-Oil Filter



Reversible - type filter for use in hydraulic systems removes foreign material down to two microns in size. Differential velocity in throat and body of venturi is utilized to force fluid through removable cartridge filter. Unit thus has no moving parts to wear and require replacement. Models are available to accommodate ¾, 1¾ and 2-inch pipe; capacities are 20, 50 and 125 gpm, respectively. Manufacturer: Machinery Equipment Div., Vickers Inc., 1400 Oakman Blvd., Detroit 32.

For further information circle MD 4 on card Page 279

## Shaded-Pole Motor

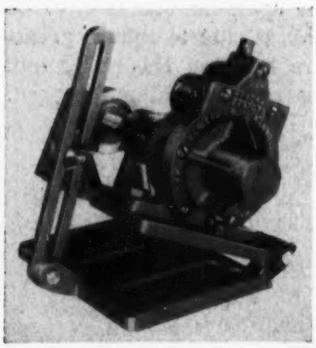


Two-pole, shaded-pole induction motor is made in ratings from 1/350 to 1/75-hp with starting torques from 1.2 to 4.5 oz-in. The motor, known as the Raytheon model 230, has been designed to combine low cost with high electrical efficiency and high starting torque. It is said to have the highest power output per pound of any unit of its size. Available in skeleton, open or totally enclosed

construction, the motor is made in types as small as $2\frac{3}{8}$ by 3 by $1\frac{3}{4}$ inches. Operating on 60-cycle current at voltages up to 250, the operating speed is 3400 rpm. Features include internal cooling fins, dynamically balanced rotor and self-aligning Oilite bearings. Manufacturer: Russell Electric Co., 340 W. Huron St., Chicago 10.

For further information circle MD 5 on card Page 279

## Tilting Pump Base



Tilting-pump base is designed for applications where it may be necessary to repeatedly disengage the belt drive. The base is bronze throughout and incorporates a double-sealed grease-packed outboard shaft bearing which retains its lubricant indefinitely and assures reduced wear on the moving parts of the pump by the extra support. Base will fit all $\frac{1}{2}$ and 1-inch single and double-impeller Eco gearless pumps. Manufacturer: Eco Engineering Co., 69 New York Ave., Newark 1, N. J.

For further information circle MD 6 on card Page 279

## Alloy-Steel Dowel Pins

New line of Unbrako dowel pins have a Rockwell C hardness of 60-62 and a tensile strength of 250,000 psi. The pins are made in a variety of lengths and in 11 diameters: $\frac{1}{8}$, 3/16, $\frac{1}{4}$, 5/16, $\frac{3}{8}$, 7/16, $\frac{1}{2}$, $\frac{5}{8}$, $\frac{3}{4}$, $\frac{7}{8}$ and 1 inch. "Red Label" type pins are made 0.0002-inch over nominal diameter, "Blue-Label" pins are made 0.001-inch oversize and "Brown-Label" dowel pins are 0.002-inch oversize. Manufacturer: Standard Pressed Steel Co., Jenkentown, Pa.

For further information circle MD 7 on card Page 279

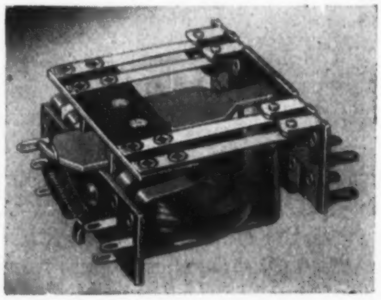
## Three-Position Relay

Polarized 3-position or null indicating relay is offered as an output device for relay-terminated control circuits and servomechanisms. The unit combines a high order of sensitivity and speed with a flexible contact structure of up to a maximum of four normally open circuits for each polarity. All eight of these circuits may carry power equal to that customarily imposed upon medium-duty relays. Using opposed windings with push-pull circuits, minimum differential power requirements are approximately 0.005-watt per contact pole. With single wound coil, about 0.0025-watt is needed per contact pole. Contacts, which may be ganged in double break or parallel-pair arrangement, are rated at 5 amperes, 110 volts, ac. Manufacturer: Sigma Instruments, Inc., 70 Ceylon St., Boston 21, Mass.

For further information circle MD 8 on card Page 279

## General-Purpose Relay

Designed for general circuit control applications, this type-C relay is made in any contact arrangement up to 4-pole double-throw. The relay can be mounted on a base or panel in any position with ter-



minals easily accessible. Average power consumption of coils is $7\frac{1}{2}$ va; contact capacity is 5 amperes at 115 volts ac. Fine silver or other specified material can be supplied as contacts. Overall size is 2 by $2\frac{3}{8}$ by $1\frac{1}{4}$ inch; weight is 3 oz. Manufacturer: Comar Electric Co., 2701 Belmont Ave., Chicago 18.

For further information circle MD 9 on card Page 279

## Hydraulic-Pneumatic Valves

Line of Unitite valves for air and hydraulic service has been redesigned to incorporate a new protective dust seal. Seal consists of a Neoprene skirt mounted on the valve stem underneath the handle, shrouding the boss on the bonnet and protecting the working parts from dust. Lubrication fitting has also

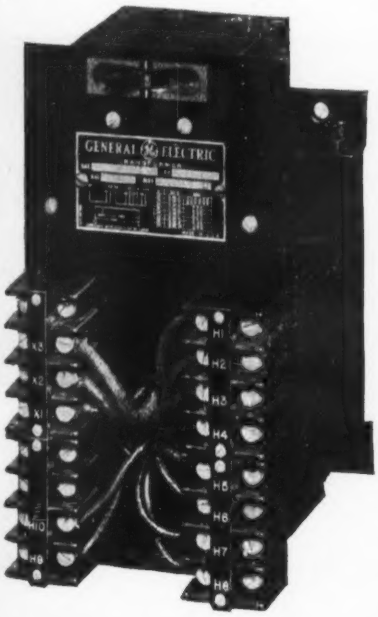
been modified. Fitting is mounted on valve stem so that the disk and stem are pressure lubricated radially outward from the center. Lubrication is accom-

plished without taking the valve out of service or shutting off flow. Manufacturer: Hanna Engineering Works, 1765 Elston Ave., Chicago 22.

For further information circle MD 10 on card Page 279

## Machine-Tool Transformer

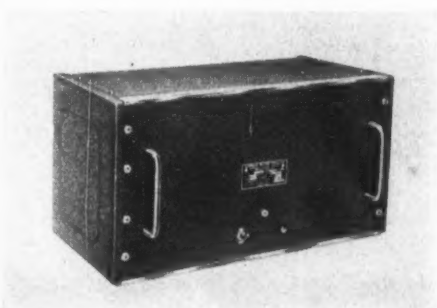
Line of machine-tool transformers is available in 25, 60 and 50-60-cycle models. For 60 cycle duty, ten ratings are made, ranging in size from 0.075 to 3 kva. They are for use across 220 or 440-volt lines. For 25-cycle applications, transformers are listed in four standard sizes ranging from 0.15 to 0.5 kva. These have primary taps for 220, 440 or 550 volts. All-purpose transformers, rated for both 50 and 60-cycle supply circuits, are equipped with 8 voltage taps in the primary, accommodating voltage range from 208 to 550. They have standard 110-volt secondaries with 92-volt taps for operation of 60-cycle relays on 50-cycle circuits. Manufacturer: General Electric Co., Schenectady 5, N. Y.




For further information circle MD 11 on card Page 279

## Voltage Regulators

Holding output voltages within $\pm 0.1$ volt of nominal value in the range 110 to 120 volts, Stabiline type IE voltage regulator is completely electronic and has



no moving parts. Unit provides instantaneous correction of line voltage fluctuations with negligible waveform distortion for input voltages from 95 to 135 volts. Performance of the new regulator is said to be unaffected by any load change from zero to full rating of the unit and independent of any load power factor change from a lagging 0.5 to a leading 0.9. Recovery time varies, depending upon line voltage, load current, load power factor and other conditions. It is, however, of the order of 3 to 6 cycles. Outstanding feature of the unit is negligible waveform distortion. At no time does the distortion exceed three per cent and for the majority of conditions of operation the waveform distortion is between one and two per cent. Voltage regulator is not frequency sensitive, operating equally well with any variation up to $\pm 10$ per cent of the designated operating frequency. It will maintain a constant voltage at any load not exceeding one kva. Manufacturer: Superior Electric Co., 580 Laurel St., Bristol, Conn.

For further information circle MD 12 on card Page 279

## Remote Indicator


Electrically operated indicator unit displays numbers from 1 to 10. Reliable operation is obtained with sequences up to 15 steps per second or a total of 0.7-second for complete step-by-step operations from digits 1 to 10. Operating on 48-volt d-c power and drawing 0.043 amperes, the unit is magnetically operated and has no mechanical attachments. Reset is accomplished by a spring return. Manufacturer: Stevens-Arnold Inc., 22 Elkins St., South Boston 27, Mass.

For further information circle MD 13 on card Page 279

## Flexible Couplings

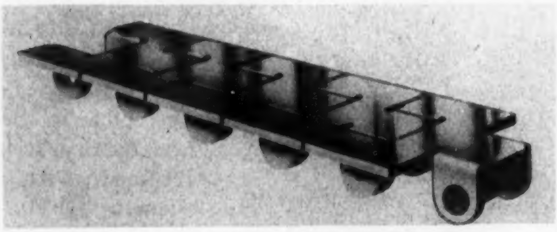
Flexible coupling using Neoprene as the resilient member is made in two styles. Type R, designed for heavy duty, has rubber core encased within steel ring which acts as compression member, taking up extrusion forces and increasing resistance to shock loads. Standard type differs from the heavy-duty type by the omission of the reinforcing ring. Both

models are available in four standard sizes: ½, ¾, 1 and 1¼-inch bores. Other bores are also available. Tolerances of the units, both concentricity and diameter, are held within +0.0005-in. and −0.0000-in. Manufacturer: Certified Flexible Couplings, 269 Lexington Ave., New York 17.

For further information circle MD 14 on card Page 279

## Conveyor Chains

New line of conveyor chains includes flat-top chain for use in conveying bottles, cans, etc. Each link of the chain is identical, eliminating need for coupling links. Top plate, which is the carrying surface, also acts as the side bar of the link, minimizing the number of parts required. The new chain is available in 1½-inch pitch and any width of top plate within the range of 3 to 7⅛ inches. It may be had in carbon



steel with pins and bushings carburized and hardened, or in type 304, 18-8 stainless steel with special alloy pins and hardened bushings. The chain will operate over No. 60 extended-pitch sprockets or over extra-width sprockets designed for the chain. Manufacturer: Wade-Morrison Co., 18401 Shaker Blvd., Cleveland 20.

For further information circle MD 15 on card Page 279

## Single-Cylinder Gasoline Engine

Air-cooled four stroke-cycle gasoline engine known as Model BB is rated at 2.5 to 4.3 horsepower. The new engine is a companion model to the Gladden 6-hp AB engine introduced earlier in the year. Measuring only 23 inches high, 15½ inches wide and 14 inches long, the power unit has moving parts largely made of aluminum alloy to reduce stress. Crankshaft is forged alloy steel, cowling is pressed steel and block is Meehanite casting. Power take-off of engine is one-inch diameter shaft. Manufacturer: Gladden Products Corp., 635 W. Colorado Blvd., Glendale, Calif.

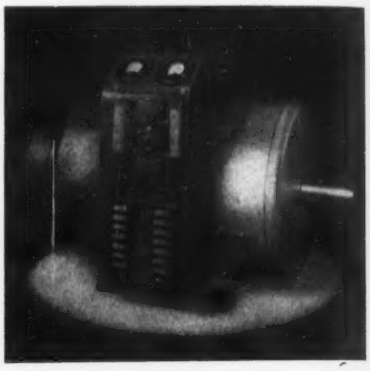
For further information circle MD 16 on card Page 279

## Flat-Head Socket Cap Screws

Addition to the Holo-Krome line of fasteners is the flat-head socket cap screw. The new screws are cold forged special-analysis alloy steel. By means of a special manufacturing method, all portions of the fastener except the threads are completely cold worked. Manufacturer: Holo-Krome Screw Corp., Hartford 10, Conn.

For further information circle MD 17 on card Page 279

## Alternator Line

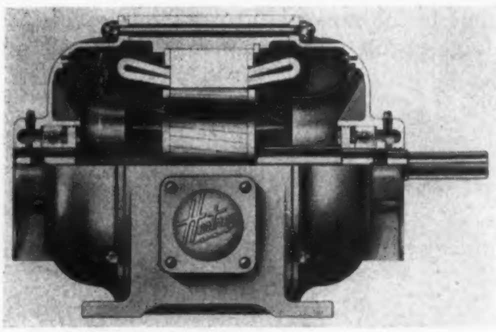


Packaged alternators for use in engine-alternator sets are compact units including revolving-field generator, direct-connected exciter, automatic voltage regulating circuit, meters and selector switch for presetting voltage. Popular ratings in 60-cycle units for use at standard voltages 80 per cent power factor include 18.7, 25, and 31.3 kva units to operate at 1200 rpm, and 25, 31.3 and 37.5 kva units to operate at 1800 rpm. Manufacturer: Electric Machinery Mfg. Co., 821 Second Ave., S. E., Minneapolis 13.

For further information circle MD 18 on card Page 279

## Polyphase Motors

Two and three-phase motors are available in ratings from 1 to 125 hp at 1800 rpm. Equivalent power ratings at other speeds are also made. Motors have



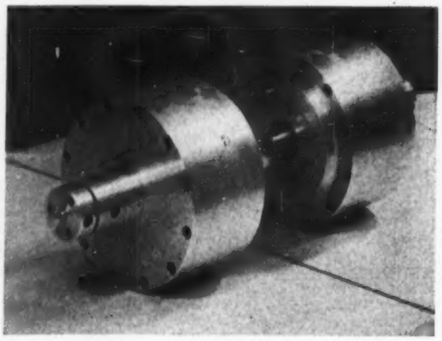
permanent-mold cast-aluminum-alloy frames, end bells and rotors. Feet and internal ribs are integrally cast. Units are designed for across-the-line starting without using reduced-voltage compensator. They operate at full rated load continuously with temperature rise of less than 40 C. Power requirements are

208, 220, 440 or 550 volts and 25, 50 or 60-cycle, 2 or 3-phase current. Manufacturer: The Hertner Electric Co., 12690 Elmwood Ave., Cleveland 11.

For further information circle MD 19 on card Page 279

## Aluminum Rotating Air Cylinders

Double-acting pneumatic cylinders for high-speed rotary applications were erroneously announced as air motors in the December issue of MACHINE DESIGN. These aluminum air cylinders are primarily recommended for use in operating rotating fixtures and



power chucks mounted on machine tools. Designed for spindle speeds up to 3000 rpm, these cylinders are available in sizes from 4½ to 12 inches in diameter. Manufacturer: Skinner Chuck Co., New Britain, Conn.

For further information circle MD 20 on card Page 279

## Variable-Speed Drives

Improved models of the Sterling Speed-Trol electric variable-speed drives feature fingertip control through entire speed range. Drive combines motor and speed control in single unit, fitting standard NEMA motor mountings. Space occupied is said to be one-half that of previous electric drives. Manufacturer: Sterling Electric Motors Inc., Telegraph Rd. and Atlantic Ave., Los Angeles 22.

For further information circle MD 21 on card Page 279

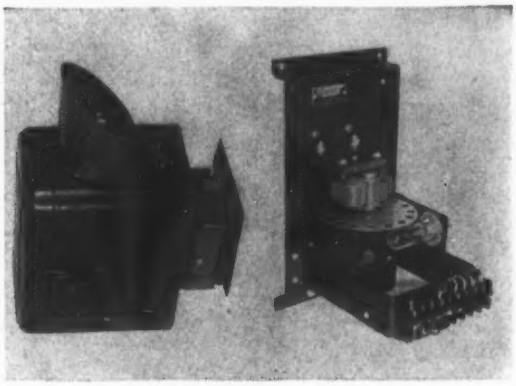
## Single-Phase Induction Motor

Capacitor-start squirrel-cage induction motor is said to have many improvements over previous motors of this type manufactured by Westinghouse. Unit is sturdier, bearings are prelubricated and windings are heavier. As a result of these improvements, starting torque characteristics have been greatly improved, and the motor requires little service. A centrifugal switch is used on motors which are rated 1 to 3 hp at 1750 rpm and also 1½ to 5 hp at 3600 rpm. On larger ratings, a newly developed relay is used. Manufacturer: Westinghouse Electric Corp., P. O. Box 2025, Buffalo 5.

For further information circle MD 22 on card Page 279

## Modulated-Light Photoelectric Relay

Photoelectric relay and light source operating on the modulated-light principle has high sensitivity. Light source contains a lamp, transformer and motor-driven slotted disk which interrupts the light beam at about 900 cycles per second. The photoelectric re-



lay has a tuned circuit which makes it responsive only to light modulated at this frequency. The unit is therefore not sensitive to changes in natural or artificial illumination. Manufacturer: Control Div., General Electric Co., Schenectady 5, N. Y.

For further information circle MD 23 on card Page 279

## Mercury Switches

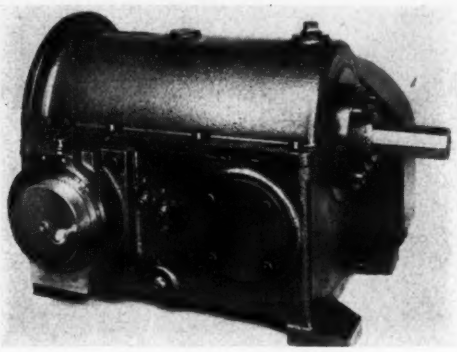
Hermetically sealed mercury switches in metal enclosures are available in ratings of 5, 10, 15 and 20 amperes. Manufactured by a method of fusing metal to ceramic, the switches are said to be especially suitable for use in explosion-proof areas, the metal enclosures providing safe and hazard-free operation in atmospheres where switch-sparking constitutes a danger. Units will operate at 110 and 220

volts ac and dc in a temperature of 100 C, can stand a 400 per cent overload for short periods. They measure $1\frac{1}{2}$ inches long and $\frac{1}{2}$-inch in diameter in the standard model, are also available in miniature size, 1 inch long and $\frac{1}{2}$-inch in diameter. Manufacturer: Mercontrol Inc., 278 Pearl St., New York 7.

For further information circle MD 24 on card Page 279

## Variable-Speed Reducer

Planetary type speed reducer, available in three models, transmits torques in the range 560 to 3800 lb-in. With input speeds of standard motors, output speeds from 6 to 220 rpm in an infinitely adjustable range may be obtained. Output of the unit is varied by means of an adjustable, speed-control belt which changes the rotation rate of the ring gear in the planetary system by means of a worm drive on the ring gear housing. Simple handwheel mechanism

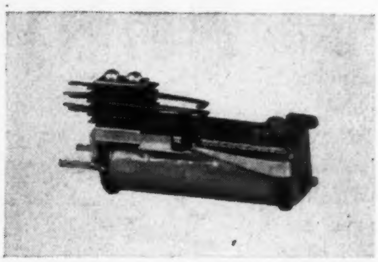


with a large, easily read dial adjusts the output-speed of the unit. Remote control by means of a variable-speed motor can also be supplied. Manufacturer: Lombard Governor Corp., Ashland, Mass.

For further information circle MD 25 on card Page 279

## Multicontact D-C Relay

Designed for use with electronic and signaling equipment, new d-c relay is capable of handling as many as 12 circuits through a wide variety of con-

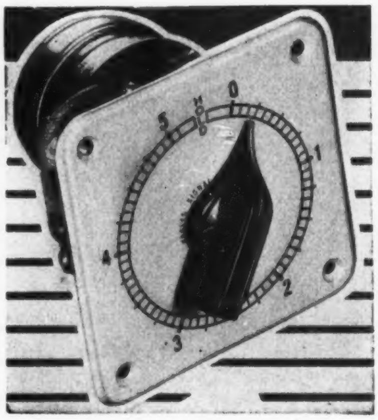


tact combinations. More than 500 different coils and five basic contact arrangements are available for use in the relay. With coil ratings ranging from 1 to 250 volts and 0.1 to 26,000 ohms, it is possible to match closely the coil voltage and resistance with the rating of the energizing circuits. Manufacturer: General Electric Co., Schenectady, 5, N. Y.

For further information circle MD 26 on card Page 279

## Manually Controlled Timer

Timer to be used on such machines as washers and extractors is synchronous-motor driven. The unit is intended to control any machine required to run a given length of time after being turned on. It is de-



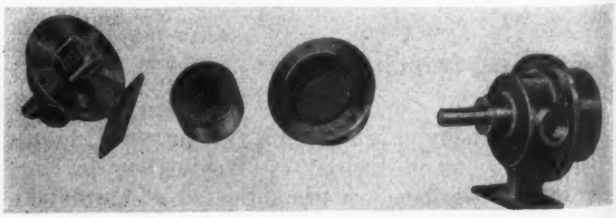

signed for flush mounting, and may be obtained with dials covering periods of time from five minutes to 10 hours. Manufacturer: Eagle Signal Corp., Moline, Ill.

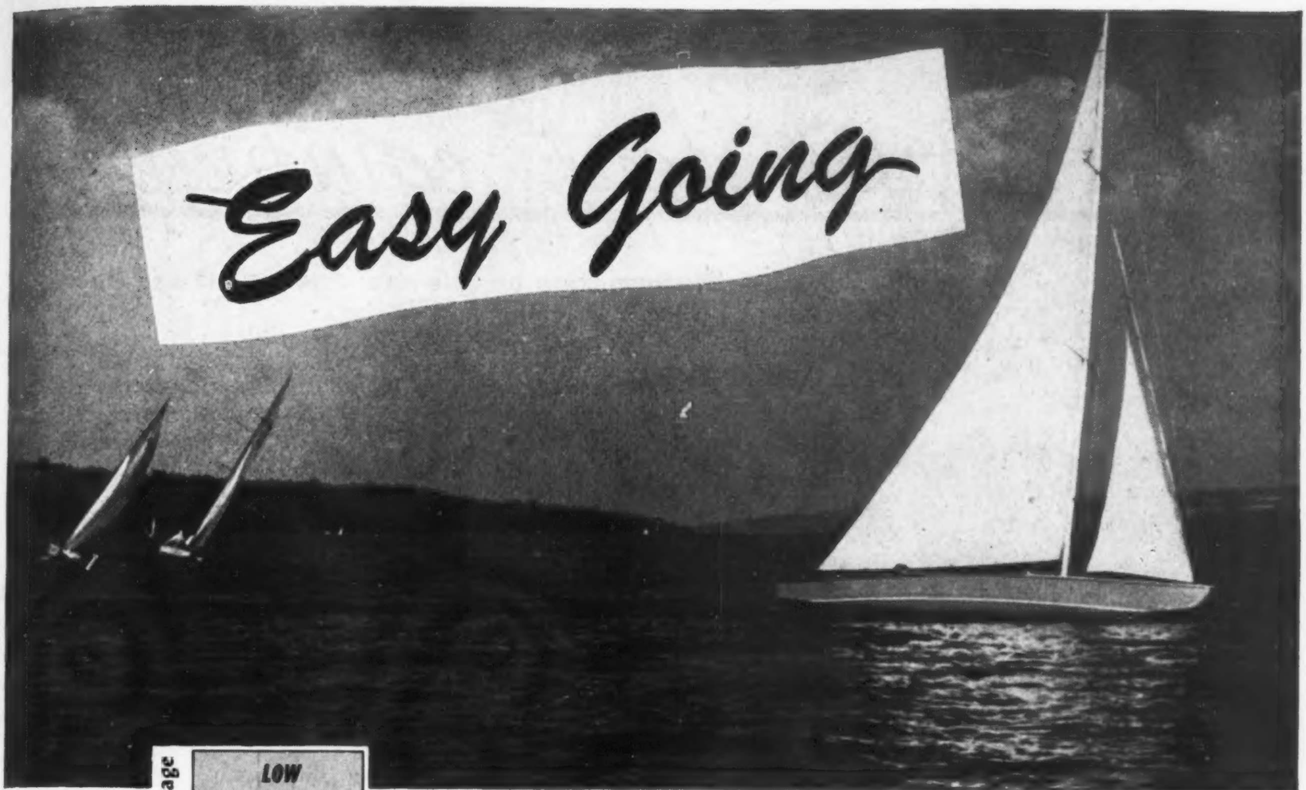
For further information circle MD 27 on card Page 279

## Rotary Fluid Pump

Rotary pump has two moving parts which rotate against flexible rubber cushions permitting abrasive materials to flow through without damage. The unit operates in either direction, is reversible and will work at any angle. It develops a constant pressure that insures a uniform flow at all speeds. Pump is made in two models, the 2-0 and the 0-5. The former has a capacity of 20 gpm operating against a 23-ft head at 1750 rpm. The model 0-5 has a capacity of 250-300 gpm operating with the same speed and head. Both types have bronze housings, mild or stainless-steel drive shafts and synthetic-rubber linings. Manufacturer: Paul Engineering Co., 11543 Conant St., Detroit 12.

For further information circle MD 28 on card Page 279



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Sleeve Type Bearings operate on practically the same principle. When the unit in which they are installed is in motion, a film of the lubricant separates the shaft from the bearing. Thus, the only friction is within this film . . . between the tiny molecules of the oil. As a result, the coefficient of friction is at the minimum virtually dependent on the viscosity of the lubricant.

Where the bearing application calls for smooth, quiet operation with a low coefficient of friction, choose Sleeve Bearings. To get the finest in Sleeve Bearings—come to Johnson Bronze. We make all types . . . we are ready to serve you now.

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**JOHNSON BRONZE**
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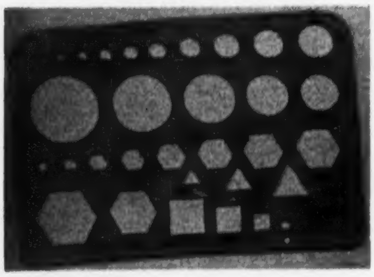
BRANCHES IN 20 INDUSTRIAL CENTERS

# engineering dept equipment

**In order to obtain additional information on this new equipment see Page 279**

### Drafting Template



Pocket-size drafting template measures 3 5/16 by 5 in. and has many commonly used geometric shapes. Size range of all shapes is 1/16 to 1 inch in basic dimension. Template is 0.030-inch thick mathematical-quality plastic with printing so applied as to not wear off. Manufacturer: Rapidesign Inc., P. O. Box 592, Glendale, Calif.

For further information circle MD 29 on card Page 279

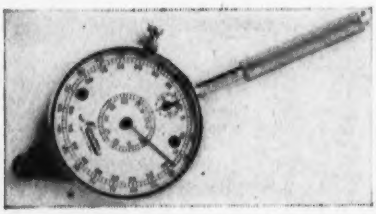
### Pencil Pointers

Pencil pointers for attachment to drafting tables or desks incorporate pencil cutting knife, pointer (either file or sandpaper) and wiping pad for removing powdered graphite from pencil point. The pointers feature handy file brush which, on some models, automatically cleans graphite from file when pointer is swung out of the way. Five models are available to suit various needs. Manufacturer: Sutz Products, 8011 S. Throop St., Chicago 20.

For further information circle MD 30 on card Page 279

### Distance-Measuring Instrument

Instrument known as the Minerva Curvimeter measures scaled distance accurately when wheel is rolled along the lines of a drawing. Three scales are incorporated in the instrument: ⅛-inch equals 1 foot,



¼-inch equals one foot and ½-inch equals 1 foot. Hands of recording dial may be reset by pushbutton. Feature of the device is that if an error has been made in measuring, the mistake can be rectified by simply reversing along the line, in which case the dial hand moves backward. Manufacturer: Herman H. Sticht Co. Inc., 27 Park Place, New York.

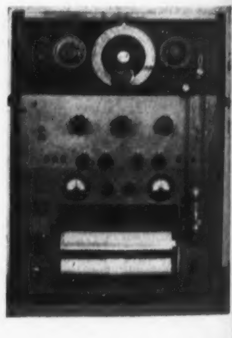
For further information circle MD 31 on card Page 279

### Chemically-Shaded Tracing Vellum

Rapid shading of tracings is achieved by the use of Craftint Doubletone tracing vellum. Tones are brought out by the application of two colorless chemicals, one chemical bringing out each type of cross hatching. The vellum is available in two types. Regular tracing vellum is used when reproduction is to be by contact printing, blue print, Ozalid or similar methods; O. S. tracing vellum is used when reproduction is to be by photography and such printing methods as offset, rotogravure or letterpress. Manufacturer: Craftint Mfg. Co., East 152nd St. at Collamer Ave., Cleveland 10.

For further information circle MD 32 on card Page 279

### Impedance-Measuring Instrument



Direct graphical recording of resistive and reactive components of impedance plotted against frequency is achieved by the Impedance Vectorgraph. Used in conjunction with a twin recorder and beat frequency oscillator, the instrument will measure resistance and reactance separately and simultaneously. It covers impedance from 1 to 4000 ohms in six ranges, with separate expansion of each scale in three fractions, ½, 1/5, or 1/10, of the other scale. Manufacturer: Sound Apparatus Co., 233 Broadway, New York 7.

For further information circle MD 33 on card Page 279

### Fluid-Flow Calculator

Slide rule for use in fluid-flow computations can be used for problems involving liquids and gases. Among values that may be calculated are orifice-plate bores, flow-nozzle dimensions and diameters of venturi throats. The instrument, known as the Flow-Rule, has a reversible slide with scales for gases on one side and liquids and steam on the other. Color is

# New Ozalid Streamliners Now Available For Immediate Delivery

**Now, you can order—and promptly receive—a new, moderately priced print-making unit that gives you these 5 new advantages at no extra cost:**

**1. SPEED.** In 25 seconds an Ozalid Streamliner reproduces your engineering drawings . . . or anything typed, drawn, printed or photographed on *translucent* paper.

**2. EFFICIENCY.** You always get an exact-size positive (not negative) copy direct from your original . . . produced in 2 quick steps—Exposure and Dry Development.

**3. ECONOMY.** An 8½" x 11" print costs you less than one and a half cents per copy.

**4. VERSATILITY.** The lines and images on your original can be reproduced in black, blue, red, sepia, yellow . . . on paper, cloth, foil, film, or plastic.

**5. SIMPLICITY.** Anyone can be the operator. Place your original on Ozalid paper and feed into the Streamliner; that's 95% of the job.

**Now an easy desk job.** You remain seated, relaxed. All controls within easy reach. Prints are delivered on top, completely dry. Another advantage: You can install your Streamliner in any drafting room or office. Only 6 square feet of floor space is required.

**A minute ago—engineering drawings.** Now she's producing beautiful Ozalid Dryphotos in seconds, in exactly the same manner. Note the size: Ozalid prints can be up to 42" wide, any length. You can reproduce advertising posters, accounting reports—the work of all departments.

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Dun & Bradstreet
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General Electric Co.
General Motors Corp.
International Harvester Co.
Lever Brothers
Montgomery Ward Co.
New York Central Railroad
Northern Pacific Railway Co.
Pan American Airways, Inc.
Paramount Pictures, Inc.
Parke, Davis & Co.
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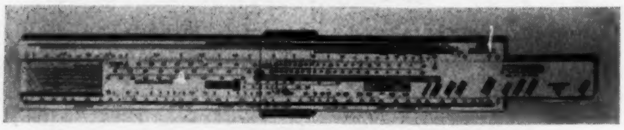
**Ozalid in Canada—Hughes Owens Co., Ltd., Montreal**



used to distinguish the scales. Rule is constructed of solid warp-proof plastic and has a heavy-leather case. Manufacturer: The Foxboro Co., Foxboro, Mass.

**For further information circle MD 34 on card Page 279**

## Sketch Paper

Drawings made on new cross-ruled sketch paper may be reproduced by such methods as blueprinting without the ruled lines reproducing. Paper is ruled in ¼-inch squares, is supplied 8½ by 11 inch in pads of 50 sheets. Manufacturer: The Ogilvie Press, 691 Fulton St., Brooklyn 17, N. Y.

**For further information circle MD 35 on card Page 279**

## Magnetic-Wire Dictating Machine

Portable magnetic-wire recording machine is designed to be used for both dictating and transcribing. Clutch built into machine permits frequent and immediate changeover from the listening position to the recording position without breaking the wire. Foot-

control switches leave hands free. Unit comes complete with one full spool of wire, one empty spool and one high-impedence dynamic microphone, either hand or desk type. Manufacturer: Peirce Wire Recorder Corp., 1328 Sherman St., Evanston, Ill.

**For further information circle MD 36 on card Page 279**

## Photographic Blueprint Paper

Silver-sensitized paper for reproducing engineering drawings on blueprint or direct-process equipment produces a high-contrast positive copy direct from positive original. The copies may be used for masters making possible quick and economical production of high quality prints. Paper is processed according to conventional practice, with the important exception that all operations may proceed in normally lighted room. Manufacturer: Eastman Kodak Co., Rochester 4, N. Y.

**For further information circle MD 37 on card Page 279**

## Flexible Curve

Instrument for use in drawing curves utilizes spring-wire forms, the shapes of which are varied by moving adjustment screws along slots. The device, known as Infinarc, is supplied together with four pre-

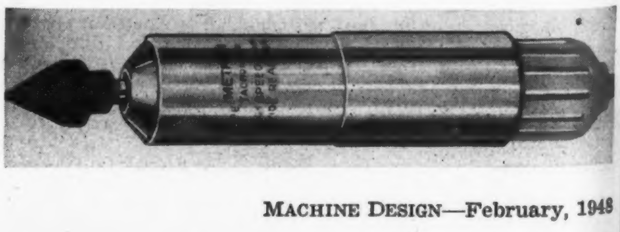
formed wire curves with which it is possible to produce almost any shape desired. Unit measures 12 inches long, is all stainless steel and comes in wooden case. Manufacturer: Cook Specialty Co., Green Lane, Pa.

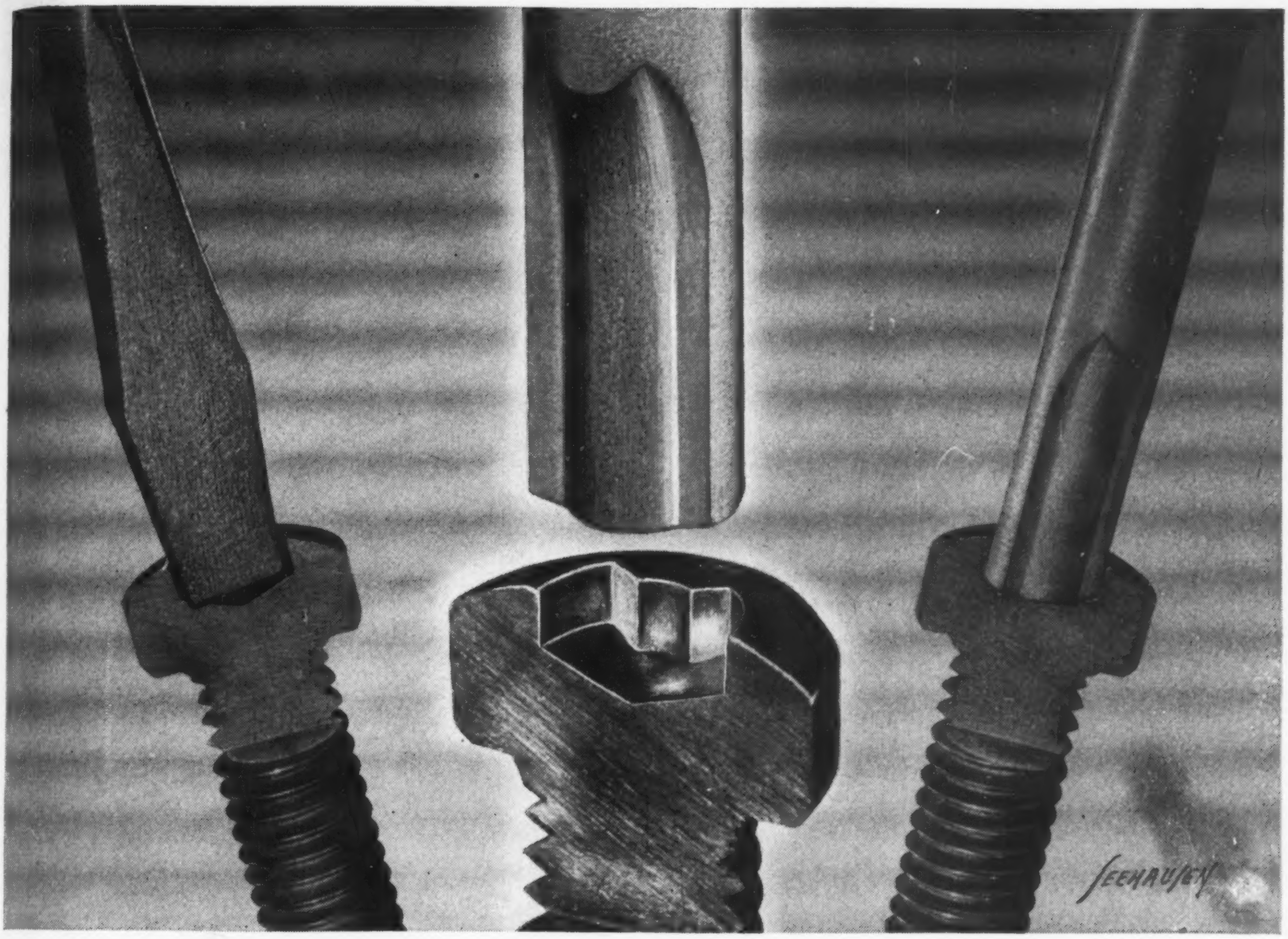
**For further information circle MD 38 on card Page 279**

## Tachometer Adapter

Adapters for use with Metron hand tachometers consist of speed reducers or increasers which fit readily over the standard tachometer head. These units consist of precision-gear speed changers with 10:1 ratios. Type 46A increases output; type 46B decreases output. Adapters have ball bearings and hardened steel gears and are designed to fit in the standard carrying case along with the tachometer itself. Manufacturer: Metron Instrument Co., 432 Lincoln St., Denver 9.

**For further information circle MD 39 on card Page 279**





# *Here's How* CLUTCH HEAD *Brings*
## *New* Safety, *New* Speed *in Line* Assembly

**Q.** What is the main cause of driver skidding? **A.** *"Ride-out" as set up by tapered driving.*

**Q.** How does CLUTCH HEAD overcome this "ride-out"? **A.** *By elimination of the tapered recess.*

**Q.** How does the CLUTCH HEAD engagement differ? **A.** *With straight sides of driver matching straight recess walls.*

**Q.** What safety benefit results from this engagement? **A.** *No slippage, so no damage to operators or work.*

**Q.** Does this eliminate need for end pressure? **A.** *Yes. No "ride-out" to combat; no end pressure; no skids.*

**Q.** Do CLUTCH HEAD users support this skid-free claim? **A.** *Many. Norge says "Cabinet damage eliminated."*

**Q.** What of this feature as a fatigue factor? **A.** *Effortless driving means more screws driven per day.*

**Q.** How does the Center Pivot Column add to safer driving? **A.** *It prevents canting by guiding bit into dead-center entry.*

**Q.** Why is CLUTCH HEAD "America's Most Modern Screw"? **A.** *Because it has features unmatched by any other screw.*

**Q.** What are these features?

**A.** *They include a recess engagement to match the ruggedness of the Type "A" Bit construction for driving up to 214,000 screws . . . non-stop; simple 60-second bit reconditioning; the Lock-On for easy one-handed driving, and basic design for common screwdriver operation.*

CLUTCH HEAD SCREWS

**Q.** And how may we check them?

**A.** *You may check all of these features by sending for package assortment of screws, sample Type "A" Bit, and illustrated Brochure. These will come to you by mail and will give you an understanding why CLUTCH HEAD users report 15% to 50% increases in assembly production.*

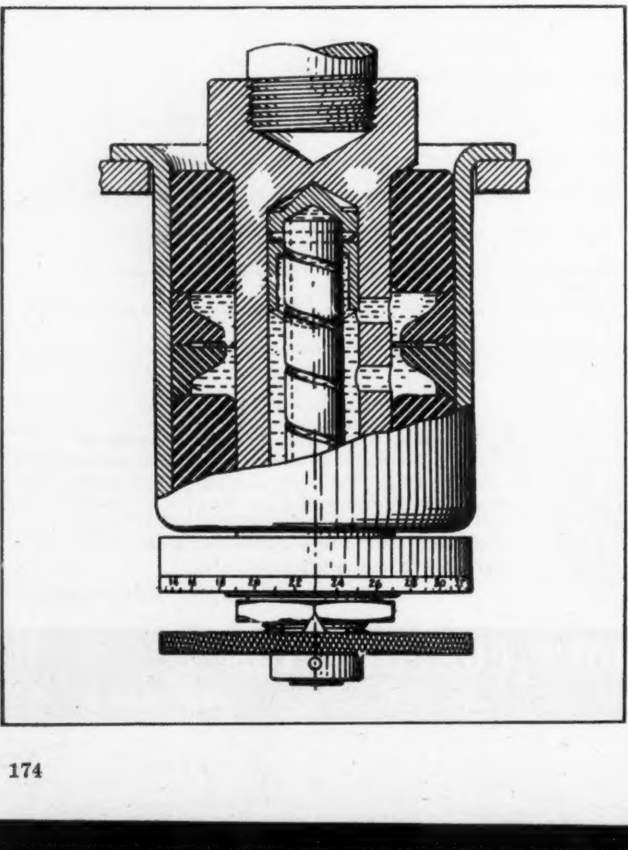
**UNITED SCREW AND BOLT CORPORATION**

CLEVELAND 2 CHICAGO 8 NEW YORK 7

# Noteworthy Patents

**OPERATION WITHOUT LUBRICANT,** a feature of unusual compressor design covered in patent 2,422,789, makes possible compression of clean, oil-free gases, liquefaction of certain gases where extremely low temperatures make lubrication impossible, compression of combustion gases where temperatures are too high for satisfactory lubrication, etc. Assigned to Arthur D. Little, Inc., by Allen Latham, Jr., the compressor has mechanically actuated admission and discharge valves which provide the most favorable operating conditions for special and unusual cases. Operable at extremely high speeds, the unit embodies a high ratio of port to piston area and employs small diameter pistons reducing leakage and sealing problems. Speed of operation also affords high efficiency and high capacity for size and weight of the unit.

**CONTROLLED DAMPING** is afforded by means of a fluid medium within the vibration absorbing mounting covered in patent 2,417,096 assigned to General Tire & Rubber Co. by L. F. Thiry. Adapted primarily for absorbing vibration by axial movement, the absorber includes means for adjusting the degree of fluid damping by hand while the mounting is in use. Included also is a bimetal element which automatically compensates for changes in damping effect of fluid caused by temperature variations.

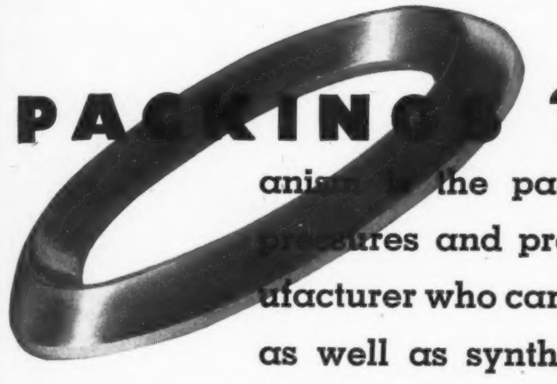


**SPEED OF OPERATION** of one or more hydraulic motors is directly indicated by the dial setting of a new hydraulic control valve the patent for which has been assigned to H-P-M Development Corp. by Johan A. Muller. Detailed in patent 2,422,119, the valve employs a tapered plug in a tapered hole to obtain extremely accurate control of the flow area and consequently accurate control of volume passed. Graduations are from zero to 9.9 and for each revolution of the handwheel the dial indexes 1/10-revolution providing settings to three figures. A carbon ring seals the control end of the valve against pressure loss and allows free rotation. Compact design of the unit allows panel mounting in a bored hole.

**A PERMANENT MAGNET** employed in an unusual gas valve covered in patent 2,417,577, gives the valve mechanism its automatic operation. Assigned to the Titan Valve and Mfg. Co. by Thomas F. Van Denberg and John Selby, the unit has an Invar push rod carried within a copper tube which expands under pilot flame heat to allow pressure on the finger pushbutton to seat the magnetic valve head in open position. Absence of pilot flame heat allows the outer copper tube to contract and this shortening action on the Invar rod breaks the permanent magnet grip to close the valve automatically under spring bias.

**UNIVERSAL HYDRAULIC CONDUIT** joint, permitting relative angular movement up to 10 degrees, is covered in patent 2,424,897 assigned to the Acrotorque Co., by Elias Orshansky, Jr. Freedom of joint operation is assured by a balanced design which prevents hydraulic reactions regardless of pressure. Antifriction bearings between the spherical surfaces of the joint portions are employed and preloading is accomplished by means of a spring.

# As a design engineer, would you like to learn more about

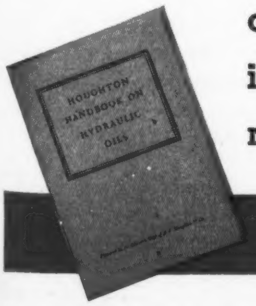


**PACKINGS?** A vital part of any hydraulic mechanism is the packing of proper type and design to hold pressures and prevent leakage. It pays to deal with a manufacturer who can provide a complete line of leather packings, as well as synthetic rubber; and who, further, can render expert engineering service on packing installations.


**HYDRAULIC OILS?** The day has long since past when just any oil can be used as the hydraulic medium. Hydraulic oils should be fortified by science to provide oxidation stability, gum solvency, anti-corrosive properties. They must have a high Viscosity Index to resist fluidity changes at varying temperatures. Engineers today recognize these "musts," and specify oils having these qualities.

## Houghton supplies, and services, both...

We are the *only* source for both packings and hydraulic oils. We've had half a century of experience in both. We render an unparalleled engineering service on both. We pioneered in additive treatment to fortify oils for this purpose. For those reasons, it will pay you to consult Houghton on hydraulics.




**HOUGHTON'S**

**HYDRO-DRIVE OILS *and* VIM & VIX-SYN PACKINGS**

For a free copy of the new Handbook on Hydraulic Oils, write E. F. Houghton & Co., 303 W. Lehigh Ave., Phila. 33, Pa.

# MEN of machines

REGINALD J. S. PIGOTT, chief engineer of Gulf Research & Development Co., has been elected president of the Society of Automotive Engineers. A member of the society since 1918, Mr. Pigott has contributed much to the organization's technical programs through both participation in the annual meetings and the preparation of numerous papers for its journal. At present he is a director of automotive engine projects for the Co-ordinating Research Council as well as being chairman of the council's engine section. He also served as an SAE councilor in 1945 and 1946. Following graduation as a mechanical engineer from Columbia University in 1906, Mr. Pigott served with various companies as chief draftsman, assistant engineer and consulting engineer. He has been responsible for the introduction of new designs in safety valves and steam gages and the design of a light-weight one-man street car with independently-sprung wheels. As professor of engineering at Columbia, he taught power station and steam turbine design and power machinery for a year. The past eighteen years Mr. Pigott has been active in petroleum research. In his present position as chief engineer with Gulf Research, he is in charge of research and design of testing apparatus and of equipment utilized in producing crude oil, both for the Gulf corporation and its subsidiaries.

*Reginald J. S. Pigott*

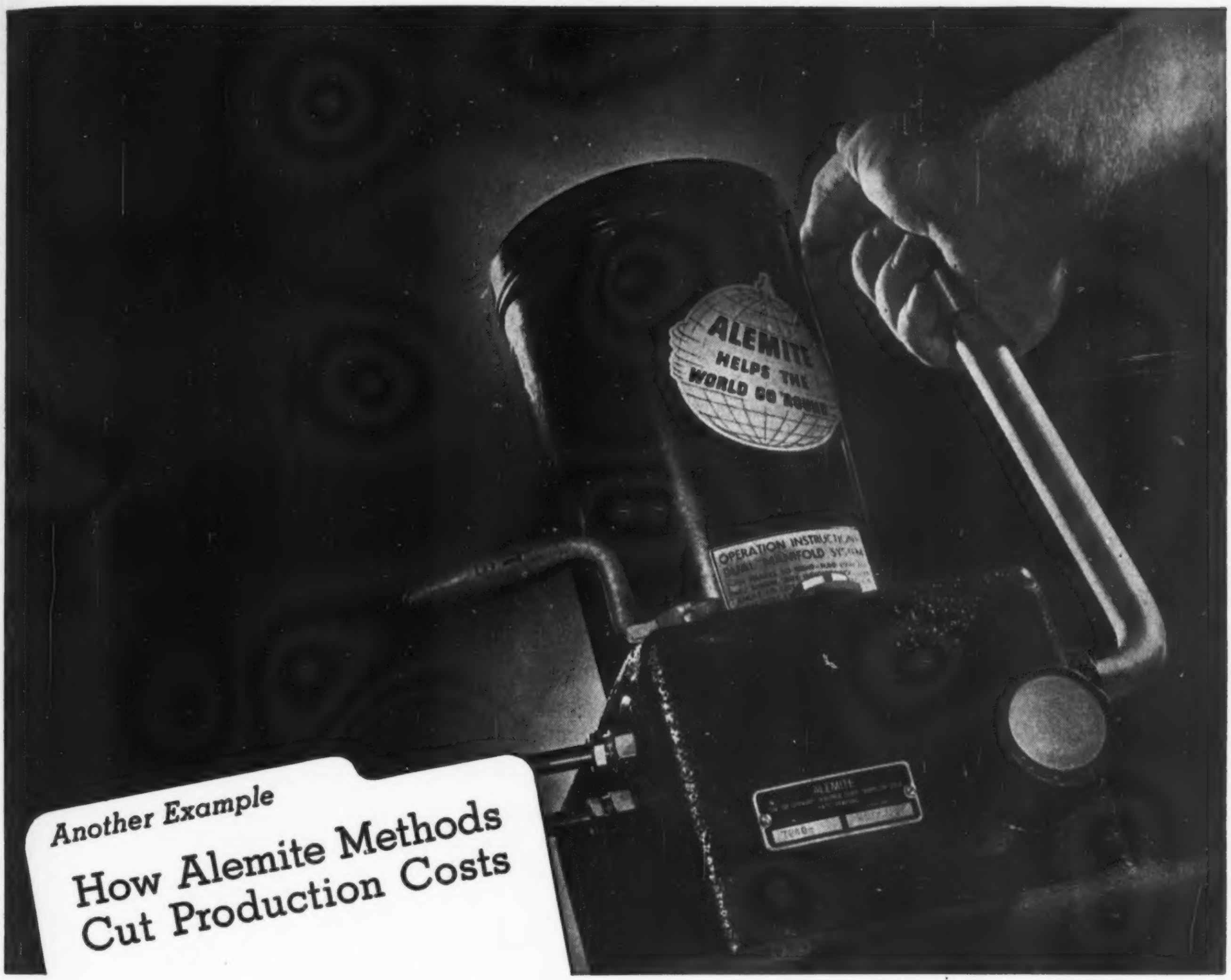
*Lovell Shockey*

LOVELL SHOCKEY, in recognition of his services in the development of many of the company's new products, has been promoted recently by the White Sewing Machine Corp. He is now supervising engineer in charge of research and production engineering as well as product and tool design. A graduate of Case Institute of Technology, 1932, he has been associated with White Sewing Machine Corp. since 1940 and his duties principally have been the supervision of tool design and production engineering. Prior to 1940 Mr. Shockey was connected with Perfection Stove Co. where he was engaged in development, research and plant engineering.

Associated with Mr. Shockey is SIDNEY HAMLETT who has been placed in active charge of research and product design. Mr. Hamlett has been with the White company since 1923 and for many years assisted CHARLES E. COLEGROVE, chief engineer for more than 50 years. Mr. Colegrove is on leave of absence but continues with the company as consulting engineer.

*Jerry M. Gruitch*

JERRY M. GRUITCH, who began his career in engineering in 1934, has been appointed director of research and development by American Car & Foundry Co. A mechanical engineering graduate of the University of Michigan, Mr. Gruitch also holds a master's degree from the Chrysler Institute of Engineering. He joined the Airtemp Division of the Chrysler Corp. in 1934 as experimental engineer, progressing to chief engineer of research and development. His work included the development and testing of heating units, air-handling systems, space coolers and large refrigerating units. In 1941 Mr. Gruitch joined the U. S.




# Reduces 182 Lubrication Points to One!

## Adds Productive Time to Machines with Faster, More Efficient Handling and Application of Lubricants

**Protect Those Machines You Design from Human Error**

Human error in lubrication can easily destroy the efficiency of the best designed machine. A five minute conference with an Alemite Representative may well reveal modern features that make lubrication foolproof. This man is fully qualified to help you because he is a specialist in modern methods of handling and applying lubricants. Write to Alemite, 1804 Diversey Parkway, Chicago 14, Illinois.

**Whether it's only one** hard to reach bearing on a machine or 182 lubrication points on another, they can all be perfectly lubricated by this modern Alemite Method. With one hand, and a few strokes on a handle, any bearing, no matter how remote, is lubricated completely in seconds from one safe, central point while the machine keeps on producing!

**An Alemite Centralized** Lubrication System eliminates "human error," adds more productive time to any machine. It delivers a measured amount of lubricant to each bearing, then signals when the job is completed. Result—no costly bearing failures due to hit-or-miss, feast-or-famine lubrication.

**Just one of many ways** that Alemite Methods help industry cut production costs through simplified lubrication procedures. And the Alemite Representative can show case after case where a time study analysis has proved that his methods reduce maintenance costs. These methods eliminate costly, time-consuming handling of oils and greases... slash shutdown time for lubrication ... completely mechanize lubrication from barrel-to-bearing ... save grease ... keep dirt and moisture out of lubricants. These benefits all add up to lower production costs and longer machinery life. Get the complete facts, today.

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**MODERN LUBRICATION METHODS THAT CUT PRODUCTION COSTS**

Army and during his four years' service was in the Ordnance Technical Research and Development section of the AAF, becoming chief of the branch in 1944. Following his separation from the Army, he served as assistant chief engineer, Dodge Division, Chrysler Corp., being engaged in the design and development of power plants and accessories. Prior to his present appointment, Mr. Gruitch was vice president in charge of engineering and a member of the board of directors, O. A. Sutton Corp., Wichita, Kans.

♦

VICE ADMIRAL GEORGE F. HUSSEY JR., USN (Ret), has been appointed administrative head of the American Standards Association. He succeeds DR. P. G. AGNEW, who for the past 28 years has served the association as secretary and head of the staff. Dr. Agnew will continue his service to the organization as consultant. CYRIL AINSWORTH, for a number of years in charge of technical activities, will serve as director of operations of the ASA staff.

♦

A. W. LEMMON has been appointed manager of research and development, chain and material handling division of the Jeffrey Mfg. Co.; J. A. FLINT, as manager of research and development of the Crusher and Traylor division; STERLING C. MOON, as manager of research and development of the mining division; and L. E. NEWELL, as superintendent of research, manufacturing and testing.

♦

SEYMOUR JEROME CHENEY before joining the Morse Chain Co., Detroit, as research engineer, had been head of the research dynamometer department of Nash-Kelvinator Corp.

♦

H. C. EDWARDS, chief engineer of research and development for Timken Roller Bearing Co., Canton, O., has been promoted to director of research and development. He has been associated with the organization since 1935 as chief engineer of diesel fuel injection equipment.

♦

RAYMOND ZANES BROWN, previously a design engineer in the Gas Turbine Laboratory of Baldwin Locomotive Works, Philadelphia, has joined the Sterling Engine Co., Buffalo, as design engineer.

♦

MORRIS G. AVERY is the new dynamometer engineer with Reo Motors Inc., Lansing, Mich. Formerly he had been associated with Continental Motors Corp., Detroit.

♦

CHADWICK S. DAUWALTER, former design engineer of Rocket Engineering Co., Los Angeles, is now in the employ of the Fairchild Engine & Airplane Corp., Oak Ridge, Tenn.

♦

SEYMOUR EPSTEIN, who had been an instructor of physics at Mohawk College, has accepted the post of instructor of design at Brooklyn College, Brooklyn, N. Y.

♦

DR. GUSTAV EGLOFF, director of research for the Universal Oil Co., Chicago, and ROBERT E. WILSON, chairman of the board of the Standard Oil Co. (Ind.), Chicago, have been added to the board of seven consultants appointed by the United States Atomic Energy Commission to speed up industrial opportunities in the field of atomic power.

♦

ALBERT C. BELL, previously designer for Cummins Engine Co. Inc., Columbus, Ind., has become supervisor of field test and design engineers for the company.

♦

RAY PERIN of Ira G. Perin Co., who for thirty-five years has been west coast representative for Elwell-Parker trucks, has been elected president of the Materials Handling Association of Northern California.

♦

R. J. SOUTHWELL has resigned from his position with American Chain & Cable Co. in order to devote his time to his new process for the manufacture of wire fabrics. He will also act as consultant for others in the development and promotion of new products and patents.

♦

W. E. DAY has been appointed director of research for Mack Trucks. In assuming his new duties, Mr. Day has relinquished the post of chief metallurgist and general foundry superintendent for this company.

♦

CALVERT CAREY and FRED DUNNING have been elected president and executive vice president, respectively, of The Yale & Towne Mfg. Co.

♦

GEORGE V. WOODLING, well known to readers of MACHINE DESIGN as author of many articles on patents, has been appointed vice president, International Electronics Corp., Cleveland, a recently incorporated firm engaged in research work on radio electronics.

♦

B. J. LEMON has retired recently after thirty-two years with the United States Rubber Co. where he was engaged in research, development and tire engineering.

♦

EDWARD C. BREINIG is no longer associated with Air Products Inc., Emmaus, Pa., as design engineer. He has joined the Aldrich Pump Co., Allentown, Pa., in a similar capacity.

♦

HERBERT C. BEHRENS has been promoted from assistant chief engineer to chief engineer, responsible for product design and development, and product standards, for Diebold Inc.

♦

CHRISTOPHER BOCKIUS, well known in the automotive field, has retired after nearly forty years in the industry. He had been connected with Chalmers Motor Co. and later the Manhattan Rubber Mfg. Co. as head of development and manufacturing methods. Five years ago he joined American Machine & Foundry Co. as director of enginering and new business research. He will continue as director of the U. S. Universal Joint Co., Detroit.

# Reversing Drives

(*Concluded from Page* 152)

prove its competitive position.

The control offered by this transmission system is probably the simplest of all the systems considered. All maneuvering is controlled by a single lever operating the pitch of the propeller blades. This provides directional control as well as speed control. However, operation at reduced powers for long periods should be obtained by changing the prime mover speed with the propeller at design pitch for best efficiency. Those installations that benefit by pilothouse control can readily be accommodated.

REVERSING GEARS: This transmission consists of a combination of gears and clutches that can provide the transfer of power with rotation in either direction from a prime mover which is not reversible. The Airflex clutch used in this arrangement can be briefly described as comprising two concentric rotating elements, one the driven member and the other the driving member. Disposed radially between the two members is an inflatable element which serves to engage the driven member when inflated.

Reduction in speed from the turbine to the propeller is accomplished by a double-reduction gear assembly. In its simplest form the turbine pinion drives one of two first gears arranged and meshed to form a gear train wherein the first driven gear rotates in one direction and the second rotates in a direction counter to that of the first. The provision of couplings makes possible the selection of direction that the propeller shall rotate and it is accomplished by inflation of one or the other of the couplings.

A 3000-hp unit represents the highest horsepower for which this type of transmission has been considered. However, studies for possible application at 6000 shaft hp indicate that no unusual difficulties need be expected. Accordingly, it would appear that the upper range of this transmission system cannot be established for the present.

Accepting the maneuverability of the present steam turbine propulsion plants as a minimum standard, the application of this transmission system should be restricted to those gas turbine arrangements in which the compressors are driven independently from the power turbine. It will be noted from *Fig.* 3 that for those gas turbine arrangements in which the power turbines also drive a compressor, the vessel will have a minimum speed in excess of 50 per cent of full speed. Further, since the torque available with this last turbine arrangement diminishes sharply with revolutions, the accelerations ahead and astern will be less than is possible with the present steam turbine drives.

Where this transmission system is used with a separate power turbine, the characteristics of the drive will be the same as the prime mover. The reversing time should be rapid and can probably be made at full revolutions if a suitable interlock is provided to reduce the fuel during the interval when one gland is emptying and the other gland is filling. With this interlock a single control lever may be used for reversing operations. Speed changes, of course, must be made by changing turbine speed.

REVERSING GAS TURBINE POWER PLANT: The possibility of achieving a satisfactory reversing gas turbine power plant appears very improbable at this time. Furthermore, there is considerable doubt that development in this direction would be warranted. Normal development of the gas turbine for use ashore will be restricted to a unidirectional gas turbine and it is apparent that all development on a reversing gas turbine must be supported by the marine application. Consequently, it does not appear sound economically to add further cost to a product that today cannot be economically justified. Further, any application of the reversing gas turbine can only be made on an arrangement containing a separate power turbine since those arrangements in which the power turbine also drives a compressor cannot produce the torque-rpm relationships necessary to reverse a ship, *Fig.* 4. This further restricts the application of the reversing gas turbine so that the development cost cannot be spread over all marine applications.

## Steam Experience of Little Value

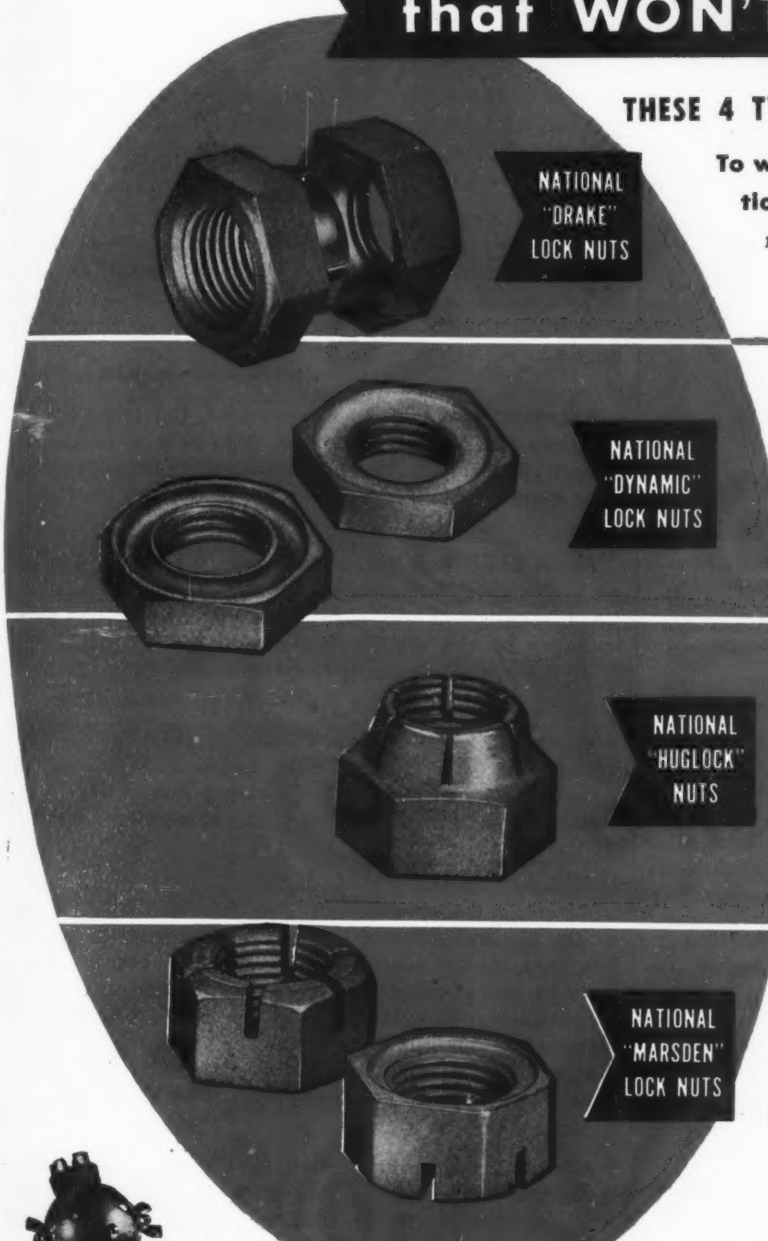
In evaluating the possibilities of developing a reversing gas turbine power plant, there is practically no experience upon which to plan other than that of the steam turbine, and several aspects of the gas turbine indicate that this experience would not be of major assistance. The fact that the windage losses in the ahead sections or elements when going astern are so much higher than those experienced in the steam turbine due to a twenty-fold increase in density, will require a new cooling concept. In addition, thermal shock of the astern turbine would have to be carefully considered in determining the rapidity at which it could be brought to power. This last item coupled with the heating of the astern elements when going ahead would indicate that the astern element would have to be maintained at operating temperature at all times by either heating or cooling as the operating conditions required. Cooling of the astern element when going ahead could be obviated by a declutching arrangement.

ECONOMIC CONSIDERATIONS: Having briefly reviewed the essential characteristics of the principal transmission systems, it is desirable to consider the economic aspects of this problem. TABLE I includes a portion of a summary of a recent economic study based upon current prices. The complete study also included drives for 3000-hp and 9000-hp installations. In order to compare the various transmissions the turbine speed was established at 3600 rpm and the propeller speed at 90 rpm.

In considering the possibilities of gas turbine power plant application, it should be remembered that for the power range presently considered, the competition is with the diesel engine having a thermal efficiency of 34 per cent. The diesel has long established its economic supremacy in the low horsepower range. Consequently, the gas turbine cannot afford to support economic parasites in the form of expensive low efficiency systems if it is to compete with the diesel.

# Self-locking LOCK NUTS

## that WON'T shake loose




**THESE 4 TYPES FIT EVERY FASTENING PROBLEM**

**To withstand severe Stress, Shock or Vibration.** Two-piece, positive lock, for use on rugged, heavy equipment, or where thickness and weight are not a factor.

---

**To withstand Shear only.** For applications requiring a thin, one-piece, light weight locking nut, and where strains would be in shear only.

---

**To withstand Shock-loading or Vibration—even under heat, oil or moisture.** One-piece, easily installed, grips the bolt threads and maintains locking effect whether seated or not. Preset torque values, built in, insure vibration-proof results.

---

**For effective locking at Minimum Cost.** One-piece, cantilever action type. Easily applied, free running until seated. Can be re-used with same locking efficiency.


**Production executives and design engineers:**


To help you select the right Lock Nut for your assembly, write today for this useful Lock Nut Booklet which gives you complete data on the engineering, design, weight and cost of these 4 "National" Lock Nuts.

National PRODUCTS

**THE NATIONAL SCREW & MFG. COMPANY, CLEVELAND 4, OHIO**

# BUSINESS AND SALES BRIEFS

APPOINTMENT of John A. Deitrich as manager of the Alloy Tube Div. has been announced by The Carpenter Steel Co. Mr. Deitrich replaces the former manager, E. W. Bachman, who will continue to work with the company in an advisory capacity.

♦

A 200 per cent increase in the production facilities for stainless-clad steel has been made by the Jessop Steel Co. of Washington, Pa. A 1000 ton per month increase in production has been achieved by the addition of new equipment and the adoption of new manufacturing techniques.

♦

According to a recent announcement, Columbus Plastic Products Co. Inc. of Columbus, Ohio is now located at 1625 W. Mound St. The new location provides increased floor space to accommodate their increased production.

♦

Saran Lined Pipe Co., a newly organized firm will act as exclusive distributors for the specialty pipe and fittings recently developed by Dow Chemical Co. Sales manager of the distributing firm is S. H. Blackmore. His offices are located in the Stephenson Bldg., Detroit.

♦

Purchase of a two-story building to augment their facilities has been announced by Leeds & Northrup Co., manufacturers of electrical measuring instruments. The building will increase effectiveness of services rendered by the company.

♦

Formerly manager of the New England sales district of the General Electric lamp department, C. C. Walker has been elected commercial vice president of the company by the board of directors. Mr. Walker will assume responsibility for customer relations in the New England territory, and will have his headquarters in Boston. T. S. Knight, whom he succeeds, retired December 31 of last year after 44 years of service with the company.

♦

Newly formed organization in the material-handling field, the Wade-Morrison Co., 18401 Shaker Blvd., Cleveland, has begun manufacture of conveyor chains. Factories are located in Columbus and Alliance, Ohio.

♦

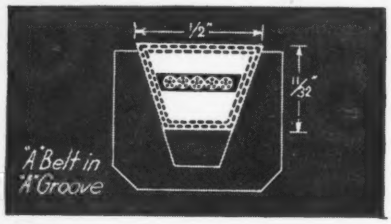
Several changes have been made in the territorial assignments of C. P. Clare & Co., manufacturers of relays. Representing the company in Ohio and Indiana are George F. Bury and H. O. Watson Jr., with offices at 508 Hippodrome Bldg., Cleveland 14. C. J. Dorr of the New York office will cover western New York state. F. O. Stratton will handle western Pennsylvania from the Philadelphia office, and E. H. Holt and W. F. Eich of the Chicago office will serve customers in Michigan and Indiana.

♦

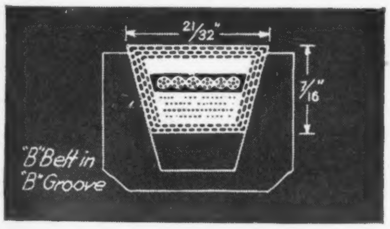
Purchase has been announced by Johns-Manville Corp. of Van Cleef Bros. Inc. of Chicago. The Chicago firm,

# FINEST LOOKING SHEAVE ON THE MARKET!


***You now have a choice of tailor-made sheaves to suit your V-Belt Drive requirements exactly!***




**"A" Groove for "A" Belt**




**"B" Groove for "B" Belt**

**GROOVED TO YOUR PREFERENCE** — an "A" belt in an "A" groove, and a "B" belt in a "B" groove—custom built—"A" for "A" and "B" for "B" . . . "it looks right"—is what you want—and you get a better-engineered, better-fitting, better-looking drive.

In addition, you get the *original* tapered cone grip sheave design—the best combination of the three basic sheave requirements. 1. Easy to Get On—2. Easy to Get Off—3. Yet Always Tight on the Shaft.

**The QD Sheave Clamps the Shaft Tighter than Any Other Sheave on the Market!**

Note, too, the famous I-Beam Construction, providing strength where strength is needed most . . . The full-size pull-up bolts . . . The taper-mated hubs and rims . . . the offset design reducing overhang stresses . . . reasons why *there's more worth in Worthington.*

Ask your nearest Worthington District Office for complete details and for assistance in designing a Multi-V-Drive to suit your requirements. *Worthington Pump and Machinery Corporation, Merchandising Division, Harrison, New Jersey. 36 District Offices throughout the U.S.*

# WORTHINGTON

# NEW!

***. . . Two More Allspeed Selectors Added by Worthington***

**YOU ASKED FOR THEM . . .**
***here they are—***

**Model D . . .** 2 to 5 hp . . . 8-1 ratio . . . speed range 360/2880 (with 1725 RPM Input)

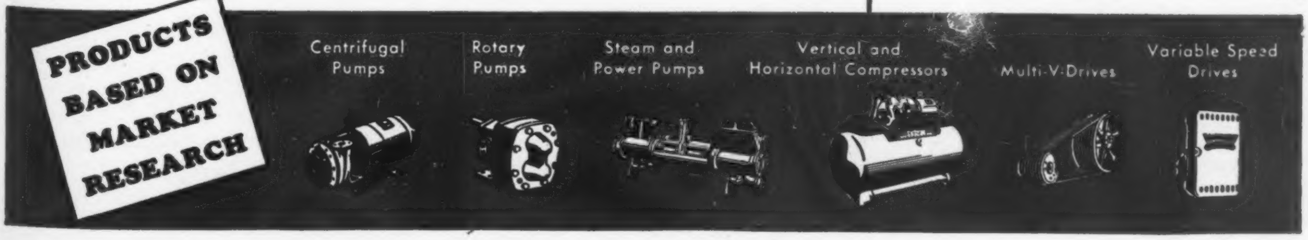
**Model E . . .** 2.8 to 7.5 hp . . . 6-1 ratio . . . speed range 400/2400 (with 1725 RPM Input)

**NOW GET ALLSPEED ADVANTAGES** in the full range from ¼ hp to 7½ hp . . . Models A and C for lower horsepowers . . . D and E for higher horsepowers. All have these outstanding features . . . Tandem belt drive for compactness . . . automatic belt tensioning . . . quick belt change . . . wide selection of speed ratios.

Ask your nearest Worthington District Office for your copy of the Worthington Allspeed Selector Bulletin, and for help in choosing the right selector for your job.

*Worthington Pump and Machinery Corporation, Merchandising Division, Harrison, New Jersey. 36 District Offices throughout the U.S.*

B7-2



manufacturers of rubber parts, will be operated as a wholly owned subsidiary of the corporation. No change is being made in the manufacturing, merchandising or distribution system that now exists.

♦

Two recently completed nonferrous foundries of the American Brake Shoe Co. were opened December 8 and 9. One plant is at Niles, Ohio; the other at Meadville, Pennsylvania. The Meadville plant will produce bronze bearings and castings, specializing in precision machine bearings of copper, brass and bronze. The Niles foundry will manufacture railroad journal bearings.

♦

Following closely the recent purchase of a German foil plant, Kaiser Aluminum officials have disclosed that The Permanente Metals Corp. is considering the addition of aluminum rod, bar and wire-making facilities at its Spokane, Washington, rolling mills.

♦

Personnel changes at the Thermoid Co., Trenton, N. J. include the following: J. Brand, formerly assistant sales manager for the automotive replacement division will handle industrial sales for the state of Colorado. He will have headquarters at Denver. J. J. Chamberlain will handle industrial sales in the state of Washington and the northern half of Oregon. In charge of industrial sales for the northern half of California and southern Oregon will be E. J. Dunlap, former head of the industrial sales promotion.

♦

Wolverine Tube Div., Calumet & Hecla Consolidated Copper Co. Inc. has opened a new mill depot on Long Island. Located at 11-26 46th Road, the depot will stock Wolverine seamless nonferrous tube in the form of S.P.S. pipe, copper water tube and refrigeration tube. G. H. Tobelman, eastern district manager, will supervise the operation from the company's office at 40 E. 42nd St., New York City.

♦

With headquarters in Houston, Texas, W. B. Whenthoff is again in charge of marketing the products of Tube Turns Inc. in that area. Mr. Whenthoff established the Houston office in 1941.

♦

Appointed assistant vice president and manager of district sales of the Vanadium Corp. of America, Harry E. Orr will be in direct charge of the Chicago sales district. Succeeding Mr. Orr as general manager of operations will be D. G. Bowser, former metallurgical assistant to the executive vice president.

♦

Announcement has been made by the General Electric Co. of four new appointments in the apparatus division's sales department. W. V. O'Brien has been named assistant manager of sales, R. M. Darrin has been made manager of central station divisions, J. C. Miller is now assistant to the manager of sales, and H. P. Bish has been appointed manager of aircraft, federal and marine divisions.

♦

A number of appointments have been made by Worcester Pressed Steel Co. Charles J. Sauter is new district agent for stampings and steel in the New York and northern New Jersey territory. Replacing G. Henry Jernberg, Mr. Sauter will have his offices at 17 E. 42nd St., New York. Representing the company in the same

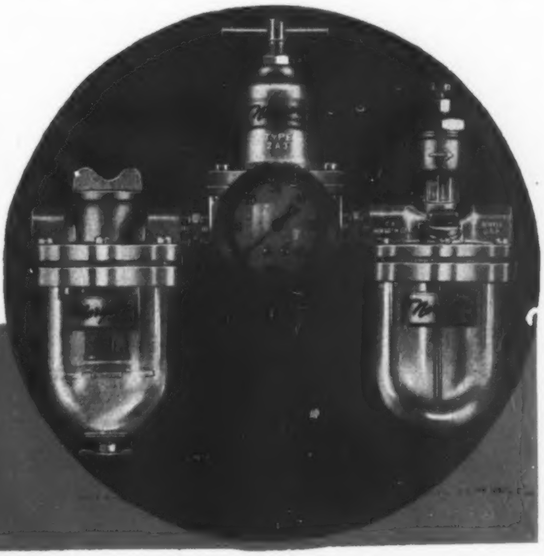
# ...way out in *Front*

**N**orgren Lubro-Control Units clean, control and lubricate the air that drives the tool or cylinder . . . for peak performance and greater production while tool operates plus complete protection against rust and corrosion while tool is idle.

Give your air cylinders and tools the punch that pays off. Investigate this compact, dependable assembly of filter, regulator and lubricator today!

**Write C. A. Norgren Co., 222 Santa Fe Drive, Denver 9, Colo.**

- **FILTER** stops rust, dirt and pipe scale, oil emulsion and moisture from reaching the tool or cylinder.
- **REGULATOR** accurately controls air power to give maximum performance with less pressure drop than conventional regulators.
- **LUBRICATOR** injects oil into the airstream, creating a protective oil-air fog. Automatic. Sight feed.



*Norgren*

capacity for the New England area will be W. Paul Moorhead. His headquarters will be at the company plant. George E. Gates, former traffic manager of the organization has been appointed assistant sales manager. Replacing Mr. Gates as traffic manager is Clifton Swanberg.

♦

Steel-distribution warehouse costing over $200,000 has been opened in the Philadelphia area by Crucible Steel Co. of America. Warehouse is located at Primos, eight miles from downtown Philadelphia.

♦

R. B. Moir, of Foote Bros. Gear and Machine Corp. has recently been advanced to the position of assistant vice president in charge of engineering and product development. His work will be with the Industrial Gear Division of the company. Also, P. H. Quackenbush, formerly assistant sales manager, will assume the full responsibility of sales manager of the industrial division.

♦

Formed from the Instrument Division of the M. B. Mfg. Co., International Instruments, Inc. of New Haven, Conn. will specialize in the production of midget meters and allied equipment.

♦

New building has been acquired by Economy Pumps, Inc of Hamilton, Ohio. Purchased from the War Assets Corp., the building contains 100,000 square feet of floor space. It will be used for the manufacture of pumps and valves by this company and its subsidiary, Klipfel Mfg. Co.

♦

Formerly general sales manager of the Firth-Sterling Steel & Carbide Corp. and founder of the Carbide Die and Mold Co., George W. Frick has been named special representative in sales development by Latrobe Electric Steel Co.

♦

Appointments to the newly created position of manager in charge of both sales and engineering have been made for all sections of the Allis-Chalmers electrical department. In this capacity for the electronics section is H. A. Bartling, formerly assistant-manager of the industrial sales department. G. W. Clothier is manager of the transformer section, and R. M. Casper is manager of the motor and generator section.

♦

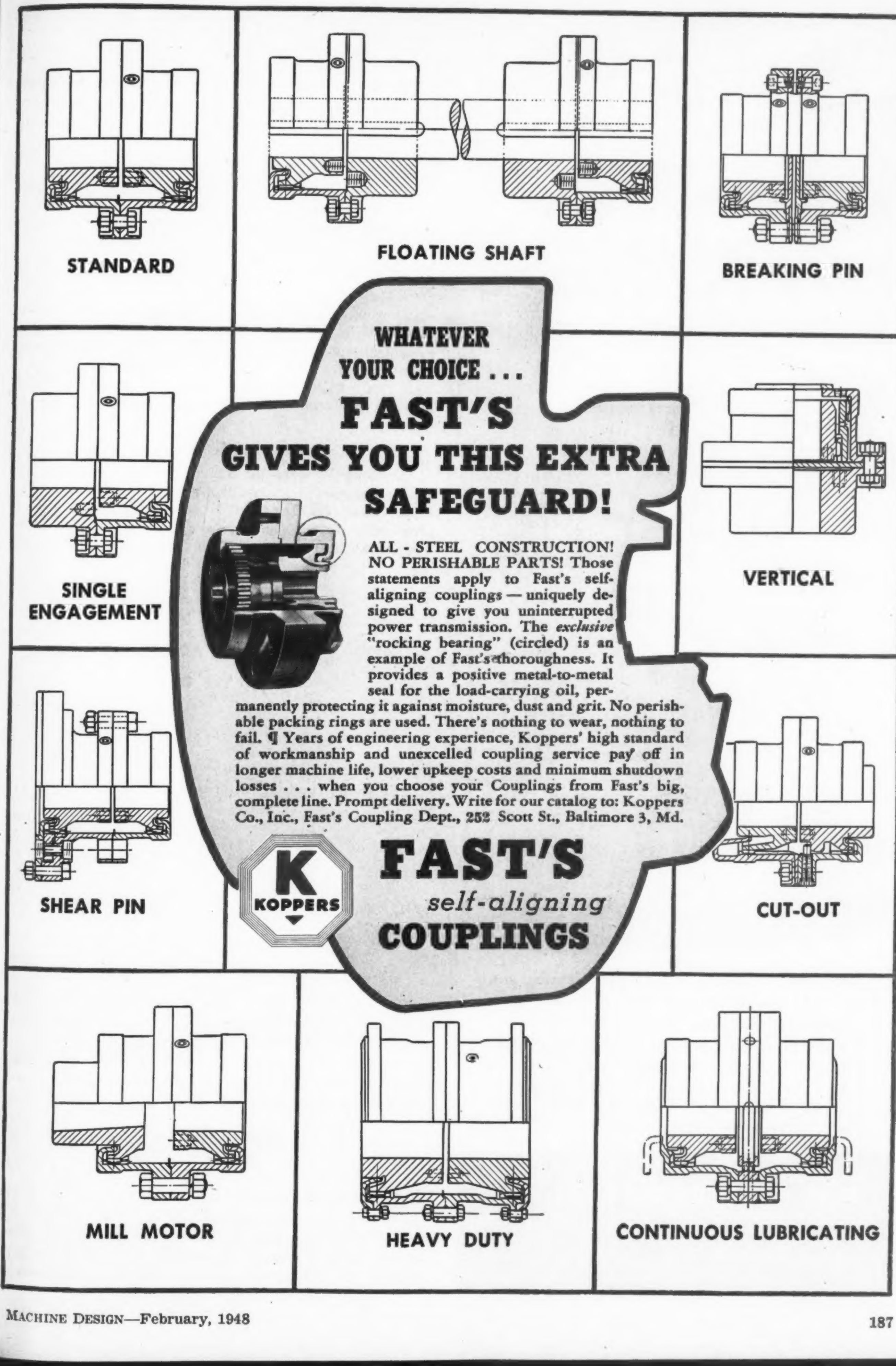
Appointed head of the Chicago district sales office of Permanente Products Co., M. D. Eisele replaces C. S. French. Mr. French will serve as district sales manager of the Cleveland office.

♦

All activities conducted by subsidiaries of the Celanese Corp. of America have been taken over by the parent company. Subsidiaries concerned include: Celanese Co. Inc.; Celanese Plastics Corp.; Celanese Chemical Corp.; Celanese Export Corp.; Tubize, Inc.; Staunton Textile Corp.; and Bridgewater Textile Corp.

♦

With the Wagner Electric Corp. since 1919 and until recently sales engineer for the company, M. E. Comstock has been appointed manager of the electrical division branch office in Boston. Another change in the Wagner sales department is the transfer of the Tulsa, Oklahoma


STANDARD
FLOATING SHAFT
BREAKING PIN
WHATEVER YOUR CHOICE . . .
FAST'S
GIVES YOU THIS EXTRA SAFEGUARD!
ALL - STEEL CONSTRUCTION! NO PERISHABLE PARTS! Those statements apply to Fast's self-aligning couplings — uniquely designed to give you uninterrupted power transmission. The *exclusive* "rocking bearing" (circled) is an example of Fast's thoroughness. It provides a positive metal-to-metal seal for the load-carrying oil, permanently protecting it against moisture, dust and grit. No perishable packing rings are used. There's nothing to wear, nothing to fail. ¶ Years of engineering experience, Koppers' high standard of workmanship and unexcelled coupling service pay off in longer machine life, lower upkeep costs and minimum shutdown losses . . . when you choose your Couplings from Fast's big, complete line. Prompt delivery. Write for our catalog to: Koppers Co., Inc., Fast's Coupling Dept., 252 Scott St., Baltimore 3, Md.
SINGLE ENGAGEMENT
VERTICAL
SHEAR PIN
CUT-OUT
K KOPPERS
FAST'S
*self-aligning*
COUPLINGS
MILL MOTOR
HEAVY DUTY
CONTINUOUS LUBRICATING

suboffice from the Houston Texas to the Kansas City territory.

♦

Douglas P. Waterhouse has been appointed northeastern district representative for General Electric accessory equipment. Mr. Waterhouse joined General Electric last year.

♦

New organization known as International Rectifier Corp. is engaged in the manufacture of colorimetric equipment, photoelectric cells and selenium rectifiers. Offices of the corporation are at 6809 S. Victoria Ave., Los Angeles 43.

♦

Robertshaw-Fulton Controls Co., with John A. Robertshaw as president, has recently been formed by merger. Organizations forming this new control-manufacturing enterprise are: Robertshaw Thermostat Co. of Youngwood, Pa.; The Fulton Sylphon Co. of Knoxville, Tenn.; and the Bridgeport Thermostat Co. Inc. of Bridgeport, Conn.

♦

According to a recent announcement, C. P. Clare & Co. has appointed Marshank Sales Co., 672 S. Lafayette Park Place, Los Angeles 5, sales representatives for California, Arizona and Nevada. Branch office of the Marshank Co. is at 1047 Flood Bldg., San Francisco.

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## Disk Stresses

(*Continued from Page* 148)

$$p=-0.724\times 65{,}400=-47{,}300 \text{ psi}$$

Again for a stationary disk, but considering an elastic shaft, the tangential bore stress from Equation 16 is,

$$q=\frac{0.022}{8.00}\times\frac{29\times 10^6}{1+0.724}=46{,}300 \text{ psi}$$

also, from Equation 11 the radial pressure at the bore is

$$p=-0.724\times 46{,}300=-33{,}500 \text{ psi}$$

For the case of the rotating disk the stress conditions can be found from the superposition of those due to fit and those due to centrifugal force. For the speed used to illustrate the tables (3987 rpm) the free bore stress is 25,700 psi. Following Hooke's law, the bore elongation, $\Delta D$, is,

$$\Delta D=\frac{25{,}700\times 8.00}{29\times 10^6}=0.0071 \text{ in.}$$

This elongation relieves the 0.022 stationary fit so that at 3987 rpm the effective fit is 0.022 − 0.0071 = 0.0149 inch. Then, for illustration using the case of the rigid shaft, the tangential bore stress is the sum of the tangential stresses due to to the centri-

# NOW YOU CAN HAVE THE PROVED ECONOMY OF A SHADED POLE MOTOR IN 1/8 HORSEPOWER!


Newest addition to the FASCO line is this Series R Fractional H.P. Motor. It is a *new, exclusive* package with definite sales advantages for you. FASCO *experienced* engineering and production know-how brings you in this FASCO shaded pole F.H.P. motor new high power, available before only in a capacitor or split-phase type motor at higher cost. Ideal for applications where starting torque is low and running torque requires up to 1/8 H.P. It is a real money-saver where continuous operation under long hour duty load is demanded. Additional advantages beside quiet, economical, trouble-free operation and dependable speed control are incorporated in this FASCO F.H.P. Motor.

## *Plus these FASCO-built quality features for better performance:*

1. INTEGRAL COOLING FAN of new design and large vent holes.
2. TAPELESS COILS, for improved heat dissipation.
3. UNIFORM ROTOR RESISTANCE and BALANCE from new pressure assembly and automatic machine welding.
4. LUBRICATED from large felt-filled reservoirs with a new oxidation-resistant oil.
5. BEARINGS are precision oil-less, self-aligning type.

***WRITE*** on your company letterhead for complete information including dimensions, specifications and performance data. No obligation.

A PRODUCT OF FASCO

F. A. SMITH MANUFACTURING CO., INC.
550 DAVIS ST. • ROCHESTER 2, N. Y.

FORMERLY *Pilot* MOTORS

fugal force and the shrink fit. This stress is then,

$$25{,}700+\frac{0.0149}{0.022}\times 65{,}400=70{,}000 \text{ psi}$$

under these same conditions then the radial stresss is,

$$-\frac{0.0149}{0.022}\times 47{,}300=-32{,}000 \text{ psi}$$

Thanks are due Mr. R. Yurtsever who helped rework the method described in K. E. Bisshopp's paper so as to have the calculation proceed from the periphery toward the bore, and for his aid in the preparation of the tabular form described.

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2. K. E. Bisshopp—"Stress Coefficients for Rotating Disks of Conical Profile", *ASME Transactions*, Vol. 66, 1946.
3. H. Haerle—"Strength of Rotating Disks", *Engineering* (London), Vol. CVI, Aug. 9, 1918, Pages 131-134.


# MEETINGS AND EXPOSITIONS

**March 1-4—**

**American Society of Mechanical Engineers.** Spring meeting to be held at the St. Charles Hotel, New Orleans. C. E. Davies, 29 West 39th St., New York 18, is secretary.

**March 3-5—**

**Society of Automotive Engineers Inc.** National passenger car and production meeting to be held at Hotel Book-Cadillac, Detroit. John A. C. Warner, 29 West 39th St., New York 18, is secretary and general manager.

**March 15-19—**

**American Society of Tool Engineers.** Sixteenth annual meeting and sixth annual industrial exposition to be held at Cleveland Public Auditorium, Cleveland. Harry E. Conrad, 1666 Penobscot Bldg., Detroit 26, is executive secretary.

**March 22-24—**

**Chicago Technical Societies Council.** Technical conference and production show to be held at Stevens Hotel, Chicago. Paul A. Jenkins, 53 West Jackson Blvd., Chicago 4, is executive secretary.

**March 30-31, April 1—**

**Society of Automotive Engineers Inc.** National transportation meeting to be held at Hotel Bellevue-Stratford, Philadelphia. John A. C. Warner, 29 West 39th St., New York 18, is secretary and general manager.

**April 5-8—**

**National Association of Corrosion Engineers.** Fourth annual conference and exhibition to be held at the Jefferson Hotel, St. Louis, Mo. W. Z. Friend, 67 Wall St., New York 5, is chairman of the publicity committee.

**April 5-8—**

**Southern Machinery and Metals Exposition Inc.** Third annual exposition to be held at Atlanta Municipal Auditorium, Atlanta, Ga. Michael F. Wiedl Jr., 267 E. Paces Ferry Rd., Atlanta 5, Ga., is managing director.

**April 26-30—**

**American Management Association.** Seventeenth packaging exposition and conference to be held at Cleveland Public Auditorium, Cleveland. Clapp & Pollak Inc., 350 Fifth Avenue, New York 1, is exposition management.

# NEW MACHINES

## And the Companies Behind Them

### Automotive

BATTERY CHARGER. Automatic; motor driven. For charging 6-cell lead-acid and 10-cell Edison batteries. Eight-hour charge period. Motor-generator and controls in cabinet 34-in. high, 24½-in. wide and 12 in. deep. Can be used on 220 or 440-volt, 3-phase, 60-cycle power supply. General Electric Co., Schenectady, N. Y.

LUBRICATORS. Cabinet types; dispense direct from 100-lb lubricant drums; hose and service units at rear. Overall size, 43¼-in. high by 18¾ x 18¾-in. Finish, baked white enamel with black, red and chrome trim. The Aro Equipment Corp., Bryan, O.

### Business

INTERCOMMUNICATION SYSTEM. Two-station—master unit and substation—with 50 ft of cable. Substation can be used as either private or nonprivate line. Power, 110-115 volts, ac-dc. Talk-A-Phone Co., Chicago 23.

DUPLICATOR. For producing copies on paper of handwritten or typed matter or drawings. Employs Ditto direct process. One turn of handle per copy; 100 copies per minute; improved liquid and pressure controls; parts easily accessible for servicing. Ditto, Inc., Chicago.

DUPLICATOR. Automatic self-feeding. Prints 2000 cards per hour. Card size, 3 x 5 in. to 4 x 6 in. The Print-O-Matic Co. Inc., Chicago 54.

BOOKKEEPING MACHINE. Portable; adding and listing, embodying addition, direct subtraction, multiplication and division. 9-column, with additional column for totals. Universal Business Machines Corp., New Haven, Mich.

### Domestic

PORTABLE ELECTRIC HEATER. Heats by radiation and convection. Uses 1250 watts on 115 volts. Rust-proof satin-finish aluminum. Weight, 5½ lb. Westinghouse Electric Corp., Pittsburgh 30.

RADIO-PHONOGRAPH. Console model. Power output, 10 watts; 12-in. electrodynamic speaker. Standard, short-wave and FM with individual built-in antennas. Eleven tubes and one rectifier; permanent-point sapphire pick-up; pushbutton tuning. RCA Victor Division, Radio Corp. of America, Camden, N. J.

CLOCK-RADIO COMBINATION. Automatically turns on radio to a set time. Combines Telechron electric alarm clock and superheterodyne receiver. Built-in antenna; case is ivory plastic. Garod Electronics Corp., Brooklyn 1, N. Y.

TABLE MODEL RADIO. Standard broadcast and FM. Continuous tone control; 4 x 6-in. permanent-magnet speaker; 7 tubes, one rectifier; 117 volts, ac-dc. Crosley Div., Avco Manufacturing Corp., Cincinnati.

### Dry Cleaning

SOLVENT RECOVERY UNIT. Recovers 90 to 96 per cent of solvent after extraction. Continuous rotating action; handles 40-lb dry load. Lint box and water separator; automatic timer; 36 x 30-in. cylinder. Vic Cleaning Machine Co., Minneapolis 3, Minn.

### Finishing

ROTO-FINISH MACHINE. For small part deburring and barrel finishing. One-compartment cylinder, 19 x 32 in. I.D., lined with replaceable kiln-dried hardwood maple. Forward and reversing switch with synchronized magnetic brake to facilitate loading and unload-

ing. Powered by 1½-hp, 220-440-volt, 60-cycle, 3-phase motor and gear reducer. The Sturgis Products Co., Sturgis, Mich.

### Heat Treating

BATCH FURNACE. Lithium atmosphere. For metal parts annealing, normalizing, hardening, spheroidizing, carburizing, carbon restoration, and brazing. Temp. range, 1200-2100 F. Conveyor and sealed chute permit direct quenching without exposure of work. Thirteen std. muffle sizes, work areas from 12 x 24 in. to 24 x 48 in. The Lithium Co., Newark 4, N. J.

RADIO-FREQUENCY GENERATOR. For high-production induction heating of parts in heat treatment. Combines 50-kw, 450-kc RF generator, matching two-position work table and built-in sink. Aluminum cabinet; complete RF shielding; stepless power output control; filtered air. Weight 7000 lb. Westinghouse Industrial Electronics Div., Baltimore 3.

### Industrial

INDUSTRIAL HUMIDIFIER. Self-contained; atomizes moisture. Vaporization capacity, over one gallon per hour; connected to any water line; directional domes. Power, 100 watts. Abbeon Supply Co., Woodside, N. Y. C.

PORTABLE SAFETY VENTILATOR. For eliminating gases, fumes, dust, etc., from inaccessible places such as within vats, sewers, etc. Electric-motor driven; capacity, 425 cu ft per min. Made of aluminum. Used as blower or exhauster. United Electric Motor Co., New York City.

GAS GENERATOR. Produces low-cost gas from fuel oil for all types of industrial ovens and furnaces. Cast aluminum chassis. Vapofier Corp., Chicago 43.

### Manufacturing

SPOT WELDING PRESS. Air or motor operated. Double-V ball-bearing type slides with adjustable ways. Capacities from 100 to 250 Kva. Eight-point selection regulator. Thomson Electric Welder Co., Lynn, Mass.

SEAM WELDER. For longitudinal or circumferential seaming. Capacities from 100 to 250 Kva with heat regulation by series-parallel connections, 8-point heat regulator, or both. Water cooled. Ball-bearing head with V-type slide. Totally-enclosed variable-speed drive at top. Thomson Electric Welder Co., Lynn, Mass.

IMPACT NUT SETTER. Electric; portable; no reaction torque. Drives at 1750 rpm; delivers 3000 impact blows per minute to tighten. Capacity, to ¾-in. nut or bolt. Reversible. Power, 110-volt, 60-cycle, ac-dc. Overall length, 12½ in.; weight, 13½ lb. Illinois Gage & Manufacturing Corp., Chicago 44.

### Materials Handling

FORK TRUCK. Collapsed height, 68 in.; max. lift, 108 in. Forks and uprights raised independently by separate rams. Full automotive-type operator controls. Automatic Transportation Co., Chicago 20.

AIR HOIST. One-ton capacity. Controlled variable speeds from creep to 17 ft per min. Standard length of lift, 8 ft. Powered by special vertical piston type air motor. Keller Tool Co., Grand Haven, Mich.

LUGGER CRANE. Mounted on half-track chassis. All-hydraulic controls from driver's seat. Lift-and-carry capacity, 6000 lb. Vertical boom travel 10 ft. Cable travel, 16 ft. Max. clearance under hook, 18 ft, 6 in. Day-Smith Hydraulic Crane Corp., Camp Hill, Penna.

### Metalworking

WELDING GENERATOR. Designed for driving by gasoline engine or power take-off. Current range, 30 to 375 amps. One control for varying welding heat. Harnischfeger Corp., Welding Div., Milwaukee 4.

DRILLING UNITS. For drilling, reaming, countersinking, spot facing and hollow milling. Two models; 6000-


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A new service, developed by the Conduit Products Division of the General Electric Company, is now being offered to design and planning engineers. If your product — in either design or production stage — has electric circuits that demand the protection of conduit, General Electric is prepared to supply your *complete wiring assembly*—a wiring harness made to your specifications, or specially designed to solve your problems.

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rpm and 9000-rpm spindles. Feed mechanism sealed in oil bath; built-in feed rate control. Adjustable stroke to ¾-in.; max. collet capacity, ¼-in. Powered by 3-phase, 60-cycle induction motors. Govro-Nelson Co., Detroit 8.

FLUTE GRINDER. For straight and spiral grinding from solid of flutes in small carbide drills, taper-pin reamers, etc., 1/16-in. diam. to 10 in. diam. Automatic indexing head. Minimum spiral lead, 5/16-in. Wardwell Mfg. Co., Cleveland 9.

CENTERING MACHINES. For centering work to be turned or ground in lathe. Single-spindle, single-head type, lever-feed type, and single-spindle duplex type. Capacities, respectively, are ¼ to 4⅛-in. diam., ¼ to 7 in. diam., and ¼ to 8 in. diam. Belt driven from built-in motors. The Whiton Machine Co., New London, Conn.

AUTOMATIC ENGRAVER. For engraving on all materials engraveable from templates or drawings. Pantograph equipped; micrometer depth adjustment. Weight, 42 lb. Airdraulics, Inc., New York, 25.

### Packaging

STAPLING MACHINES. For closing tops and bottoms of corrugated or fiber cartons simultaneously from outside. Retractable anvil drives staple from outside and clinches on inside. International Staple & Machine Co., Havertown, Pa.

FILLING MACHINES. Automatic auger type, developed for small packers. Packages coffee, spices, cocoa, etc. Filling range, 1 oz to 5 lb; filling and weighing-speed capacity to 30 packages per minute. Triangle Package Machinery Co., Chicago.

### Photographic

ROLL-FILM DEVELOPING UNIT. Portable, motor-driven; handles 35-mm roll film up to 100 ft long. Automatic or manual operation. Made of stainless steel and plastics. Solution required, 3½ quarts. Power, 110 volts, single-phase, a-c. Weight, including case, 14 lb. Fairchild Camera and Instrument Corp., Jamaica 1, N. Y.

MOVIE CAMERA. 16-mm semiprofessional. Professional shift-over focusing; four-lens turret; viewfinder parallax adjustment; positive viewfinders; 400-ft capacity. Bell & Howell Co., Chicago 45.

### Plastics

INJECTION CYLINDER. For all thermoplastics. Independent temperature control of spreader and cylinder body by pyrometers. Exposed surfaces polished and chrome plated. Lester-Phoenix, Inc., Cleveland.

### Processing

GRIZZLY FEEDER. Double-magnet, heavy-duty electric, vibratory. For large-capacity separation of crusher feeds. Variable control of feed rate. Power, 220 to 440-volt, ac. Syntron Co., Homer City, Pa.

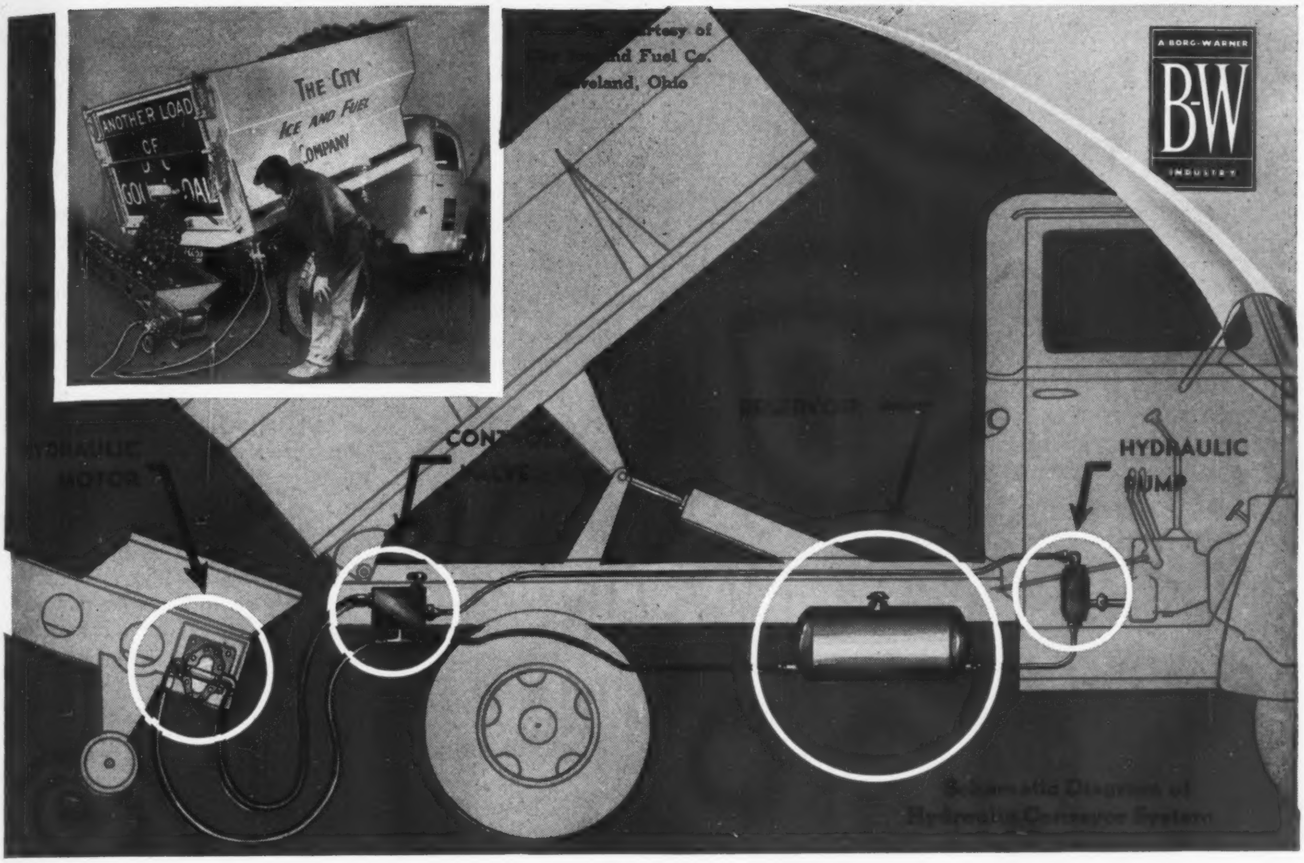
STIRRING MIXER. For all fluids; handles containers from one quart to five gallons; motor bracket adjustable in height and swiveled to allow 90-degree vertical swing of mixer. All parts rustproofed. Power, 110-volt, 60-cycle. Weight, 10 lb. The Dayton Rogers Manufacturing Co., Minneapolis 7.

### Road Machinery

EARTH MOVER. High-speed, electric controlled; 35-ton capacity. Powered by 225 hp diesel. R. G. LeTourneau Inc., Peoria, Ill.

TANDEM ROLLER. 3 to 4-ton type. Simple hand clutch lever provides forward or backward movement. Fingertip starting with handwheel or hydraulic lever. Wheeler Div. of Shaw Sales & Service Co., Los Angeles.

ROAD-SWEEP MAGNET ON HOIST TRUCK. For picking up scrap metal from roadways. Power for the 18 x 96-in. electromagnet is provided by gasoline-motored generator. Stearns Magnetic Mfg. Co., Milwaukee 4, Wis.



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# DESIGN ABSTRACTS

## Ceramics for Turbine Blades

SERVICE requirements in a jet motor are severe because they involve conditions of high tensile stress, as well as high temperature, for prolonged periods of time. Superimposed upon demands for high mechanical strength at temperatures in excess of 1200 F are the requirements for resistance to severe fluctuations in temperature and to some degree of vibrational stress. Assuming a specific gravity of 8.3 (typical for a high-temperature metallic alloy), one can calculate a possible maximum root stress of at least 40,000 psi in the blade of a modern turbojet motor. A number of metallic alloys have 100-hour stress-rupture tensile strengths in excess of this value at 1200 F—a greater proven tensile strength than any known ceramic body possesses at any temperature—but the strength of metals will, in general, decrease rapidly above 1200 F. In comparison, refractory porcelains have been developed that retain at elevated temperatures much of their initial, though lower, strength. Some of these porcelains have sustained stresses in tension of as much as 17,000 psi at 1800 F, for periods exceeding 100 hours, and with an average elongation, or creep, of less than 0.0004 per cent per hour.

### Low Density Reduces Stress

The ceramic has the added advantage of a low density when compared to metallic alloys. Stress being directly proportional to the density, a porcelain blade having, for example, a specific gravity of 3.0 and a tensile strength of 17,000 psi, would be the equivalent, in strength, of a metal with a specific gravity of 8.3 and a strength of 47,000 psi. At 1800 F, however, the best alloy probably has a 100-hour stress rupture value of less than 12,000 psi.

Another characteristic of porcelains is that they do not go into the so-called third stage of creep in which the rate of creep increases with time. Rather, under prolonged loading, they appear to become more resistant to creep. The picture is not without its flaws, however. It still remains to be determined whether or not these porcelains, or others to be developed, will have the necessary resistance to thermal and mechanical shock required for turbine blade service. Even though it should be found

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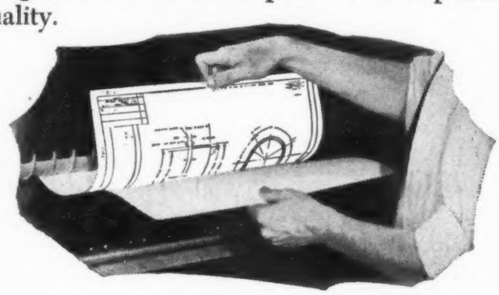

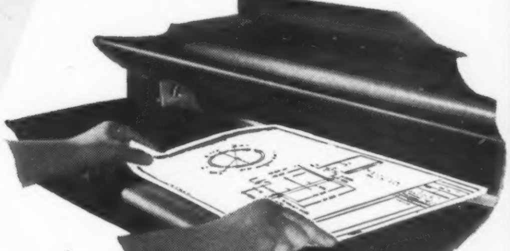

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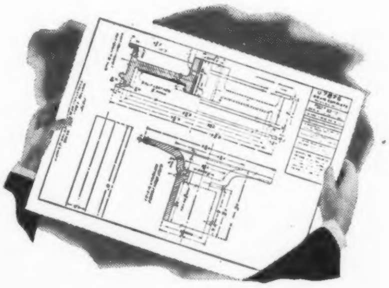

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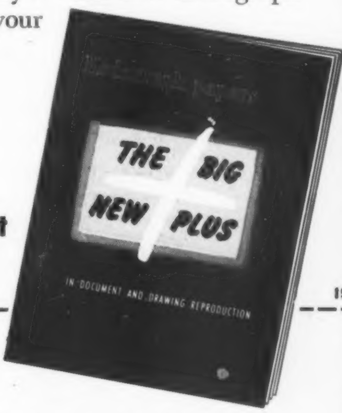


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that they do not, it is practically inevitable that they will replace metals in other high-temperature applications now being considered by engineers.—*From a paper by R. F. Giller, Chief, Porcelain and Pottery Section, National Bureau of Standards, presented at the recent annual meetig of the SAE in Detroit.*

## Materials and Design

IN THE race for acceptability of the diesel engine the future will largely be determined by our ability to utilize materials effectively—not only the materials which we have at hand but also the super materials in various stages of development today. At times we bemoan our lack of knowledge of combustion phenomena but at that our experience and ability to utilize combustion effectively are far ahead of our knowledge of the utilization of the materials and their economic evaluation. Hand in hand with combustion studies go the material structure requirements in order to provide light weight and suitable endurance life under the operating conditions of high outputs and higher speed. These facts have been well understood in the aircraft industry but are relatively new in the diesel engine industry. A new field has opened up for stress study and fatigue phenomena knowledge because of the specific operating characteristics of the diesel, making somewhat different demands on materials than are experienced usually in aircraft industry. We still bear the yoke of weight in our design, as pertaining to our traditional concepts of the diesel engine.

### Fatigue Stress Factors

Improved knowledge of stress analysis and fatigue studies has fostered the development of the geometry of structures to the extent that the life of component parts is lengthened and better balanced, resulting in longer operating period without shutdown or costly maintenance. Studies in fatigue and stressing factors and more careful examination of the factors influencing vibration have been given significant attention. Whereever outputs are increased and energy levels raised, there is the opportunity for weight reduction per bhp and the effective utilization of higher speeds. All of these factors have required study of the vibration problem. Whereas at one time little attention was given to the noise levels inherent in the operation of diesel engines,

the trends of development have necessitated the closer scrutiny of vibration problems as to their effect on noise. It is perhaps on the one hand an admission, yet on the other hand it is a measure of achievement to state that the diesel engine has acquired smoothness of operation in high speed units quite comparable to that customary in gasoline engines.—*From a paper by C. G. A. Rosen, director of research, Caterpillar Tractor Co., presented at the recent Midwest Power Conference in Chicago.*

## Silicone Lubricants

USING silicone fluids, journal bearings may now be operated at higher temperatures than has been the practice with petroleum oils. In investigations it has been found that up to 300 to 375 F the fluids may be used without significant deterioration for long periods of time, although above 400 F the rate of thickening, discoloring and precipitation of gel increases rapidly, indicating need for periodic inspection or cleaning. The change in viscosity of the fluid at a given operating temperature is practically linear after the first three hours when using dimethyl fluid and after the first 40 hours when using the methyl-phenyl silicone fluid. At bearing temperatures of 450 to 500 F in such systems the dimethyl silicone fluid cannot be used for more than approximately 30 hours unless the lines feeding the bearings are blown free of restrictions due to the gelled oxidation products. Therefore, for continuous hot operation over hundreds of hours or more the bearing temperature should not exceed 400 F.

The dimethyl silicone fluids have low pour points (from −65 F to below −100 F) in the low viscosity grades, while the methyl-phenyl silicones have higher pour points (about −5 F). However, the latter are the more resistant to viscosity changes wrought by high temperature and oxidation. Both fluids are more stable than nonadditive petroleum oils at the same temperature. The ASTM viscosity-temperature slopes of the dimethyl silicone fluids are always less than those of methyl-phenyl silicones having the same viscosities at the reference temperature. Both classes of silicone fluids have ASTM slopes less than those of comparable petroleum oils. This permits one viscosity grade of silicone fluid to be used over a wider temperature range than is permissible with a single petroleum oil. The low

# WHAT HAVE ALUMINUM EXTRUSIONS FOR TEXTILE MACHINERY GOT TO DO WITH MY PROBLEM?

ARE you interested in cutting machining costs, reducing the weight of a moving part without sacrificing strength, and adding service life to a part or a product? If so, aluminum extrusions do have a bearing on your problem.

Expensive wooden handrails on looms—cast iron ring rails, spindle rails and roller beams for spinning frames—all are being replaced rapidly by parts extruded from high strength, heat-treated Reynolds Aluminum. Why? Because aluminum handrails are not affected by humidity, will not distort. Use of extrusions in place of cast iron parts cuts machining costs to the bone. Light weight aluminum reduces inertia in reversing motions, cuts down vibration. Freedom from rust means elimination of protective coatings, less maintenance.


*The base price of aluminum has been reduced 30% since Reynolds became a primary producer.*

Can you afford to use aluminum? Remember, every pound offers 3 times the working area of steel, brass or copper. Now add the economy of handling and shipping this lightweight material plus its functional advantages. You may find that aluminum is one of today's best bargains. A Reynolds technician will be glad to assist you in such an analysis. Just contact your nearest Reynolds Field Office or write to Reynolds Metals Company, 2521 South Third Street, Louisville 1, Kentucky.

**REYNOLDS *Lifetime* ALUMINUM**

surface tensions and creeping properties of the silicone fluids causes greater sealing difficulties than are encountered with petroleum oils.—*From a paper by J. E. Brophy and J. Larson, U. S. Naval Research Laboratory, and R. O. Militz, R. T. French Co., presented at the recent ASME Annual Meeting in Atlantic City.*

## How Light Is Light?

THERE is no particular magic about making a car light in weight. This can be done by simply leaving things off and using thinner sections. The real technique comes in making the car light and at the same time of acceptable appearance with proper performance, satisfactory road-handling characteristics, and adequate durability.

In tackling this problem in the United States the goal must be set that the vehicle not be small to the extent of inviting ridicule or of crowding the passengers, as is the case in some foreign cars. If at all possible, it should appear and perform in all respects like the cars which the public is accustomed to driving.

### Weight and Cost Reduction

There are various methods whereby weight and cost can be eliminated, the principal ones being the following:

1. *Making the vehicle smaller.* This is more effective in weight than in cost reduction, since the main saving is in raw material. The cost of labor and tooling are not greatly affected.

2. *Elimination of unnecessary parts.* This shows savings in weight as well as in material, labor and burden without affecting the commercial appeal. It is highly recommended.

3. *Simplification of design.* By this is meant the elimination of unnecessary machining operations. While weight may not be affected it is quite important from the standpoint of reducing the cost of labor as well as investment in tools and machinery. Here again commercial acceptability should not be impaired.

4. *Combining several functions into a single part.* This is really a combination of items 2 and 3, a good example of which is the rear trunk lid handle, lock and license-plate lamp used on some cars. Weight and cost are reduced while commercial acceptance may even be enhanced.

5. *Using higher stresses.* This is a good practice provided stresses remain within safe limits. If offers a

# If you make portable equipment designed to save man power—

Does portability sell your product? Take a good look at Magnesium. You'll find that it can give you *greater portability*—at costs that are in line with the competitive advantages gained. In a product such as yours, Magnesium's lightness, strength and versatile workability can pay off—in increased serviceability and widespread acceptance. Dow, as a major producer of ingots, castings, sheet, plate, and extrusions is always ready to assist manufacturers in adapting magnesium to their problems.

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# Magnesium pays!

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*Problem*—Maintaining an uninterrupted flow of material and stock throughout the plant to keep production lines going. *Solution*—a Magnesium hand truck that is easy to move—cuts operator fatigue. Simplified Magnesium design saves manufacturing costs. Body: Sturdy aircraft magnesium alloy extrusions combined with a cast toe-plate. Ready for all around duty—yet weighs only 10 pounds.

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*Problem*—Expediting loading operations—requires a dockboard that can take heavy loads yet be moved quickly from car to car. *Solution*—Magnesium does the job. This dockboard is 42 by 66 inches . . . weighs only 77 pounds . . . carries a load of 1,600 pounds. Built of quarter-inch plate, reinforced with 2-inch I sections and with side pieces of 3-inch channels—all welded construction. Another example of Magnesium combining lightness and strength to save man power.

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*Problem*—Getting more production by making this powerful saw more portable—easy to move from job to job. *Solution*—Magnesium's versatile design properties made it possible. This manufacturer achieved the strength and durability he needed . . . and saved 36 pounds. Sturdy Magnesium castings form the motor housing and back handle weighing only 2½ lbs.; gear housing 3½ lbs.; upper guard 2 lbs.; cover guard 1 lb.; front handle ¼ lb. and bearing cap ½ lb.

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saving in weight and material cost without affecting other characteristics.

6. *Using light metal alloys.* This is the most publicized solution to the problem probably because it can be ordered by telephone. It allows a reduction in weight but at an increase in material cost. The use of lightweight alloys usually requires a new technique which is not commonly used in the motor car industry and involves further increase in cost to cater to the different characteristics of the material. It is most useful when applied to income-producing vehicles such as motor busses, but in privately owned passenger cars the increased cost can rarely be amortized by the additional economy of operation within the time the average first buyer owns the car.

### Sacrificing Appearance

If a lightweight vehicle is to be successfully achieved the aesthetic must take second place to the functional. Weight must be taken out of all possible units. Ounces must be eliminated from light parts and pounds from heavy parts. Most of the potential weight saving will probably come from the body because it is not only the heaviest single unit but is, relatively, not highly stressed as a structure. It contains a high percentage of nonstructural material in the form of trim and garnishing, which has a considerable latitude in its commercial appeal.

Finally, every pound of nonstructural or parasite weight removed from the body will allow a reduction in weight of those units which serve to support the body as well as to start, stop and steer the vehicle. It is safe to say that for every pound of parasite weight removed another pound of chassis weight can be saved.

To provide transportation in which the cost of the vehicle, as well as the operation and maintenance, falls within the means of millions in this country we must build lighter, more efficient, and more economical cars. If the American automotive industry is to retain the dominant position in world trade which it has held for many years, and if we are to avoid losing our foreign markets and prevent the flooding of our own shores with economical transportation built in foreign countries at lower labor rates, we must do so by building a strong position in our own country.—*From a paper by W. D. Appel, Willys-Overland Motors Inc., presented at the 1947 annual meeting of SAE in Detroit.*